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Nov. 19, 1889—EIFFEL, GUSTAVE.............37 Rue Pasquier, Paris, France.

Nov. 6, 1884—CRASHOF, F..........................Carlsruhe, Germany.

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April 7, 1880— }
May 14, 1890— } PORTER, CHAS. T.*...............Box 79, Montclair, N. J.

Nov. 2, 1882—REED, SIR EDWARD J., K. C. B., M. P..Broadway Chambers Westminster, S. W., London, England.

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May 26, 1886—WALKER, FRANCIS A....President Massachusetts Institute of Technology, Boston, Mass.

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* Manager 1882-85; Vice-President 1886 88.

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June 20, 1880—NORMAN, GEORGE H............343 Beacon St., Boston, Mass.
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June 23, 1880—ALLEN, JOHN F............629 Walton Ave., New York City.
May 21, 1884—ALLISON, ROBERT......Franklin Iron Works, Port Carbon, Pa.
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Nov. 29, 1887—AMES, WILLIAM LEWIS....Prof. Mech. D'w'g, Rose Polyt. Inst., Terre Haute, Ind.
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May 27, 1885—ASHWORTH, DANIEL....Mech. and Cons. Eng., Box 72, and 349 Edwin St., E. E., Pittsburgh, Pa.
Nov. 29, 1887—ATKINS, JAMES.............................Montclair, N. J.
Aug. 10, 1881—AUCHINCLOSS, WILLIAM S....209 Church St., Philadelphia, Pa.

Oct. 16, 1888—BABBITT, GEORGE RODNEY..Supt. W. A. Harris Steam Eng. Co., Providence, R. I.
April 7, 1880—BABCOCK, GEORGE H..Pres. The Babcock & Wilcox Co., 30 Cortlandt St., N. Y. City, and 17 West 8th St., Plainfield, N. J.
May 4, 1881—BABCOCK, STEPHEN E...Chief Eng., W. W., and 524 Monroe St., Little Falls, N. Y.
May 15, 1889—BACKSTROM, GUSTAF LEONARD..Mech. Eng., 1600 Hamilton St., and 1609 Summer St., Philadelphia, Pa.
Nov. 11, 1885 } May 26, 1886 } BACON, EARLE C..Copeland & Bacon, Mech. Eng., 85 Liberty St., New York City, and 52 Ft. Greene Pl., Brooklyn, N. Y.
Nov. 6, 1884—BAILEY, READE W.....Mgr. The Robinson Rea Mfg. Co., 53 Carson St., and Bailey Ave., Pittsburgh, Pa.
April 7, 1880—BAKER, W. S. G.*..Pres. and Treas. Balto. Car Wheel Co., P. O. Box 176, Baltimore, Md.
May 14, 1890—BALDWIN, BERT. L.....Mech. Eng., Room 20, Lincoln Inn Court, and 1009 Gilbert Ave., Cin., O.
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April 7, 1880—BALDWIN, STEPHEN W.†..Penn. Steel Co., 2 Wall St., and 267 West 73d St., New York City.
Nov. 2, 1882—BALDWIN, WILLIAM J....M. E. Heating and Ventilating Co., 277 Pearl St., New York City, and 1047 Herkimer St., Brooklyn, N. Y.
Nov. 1, 1883—BALL, FRANK H.‡....Treas. and Gen. Mgr., Ball Engine Co., Erie, Pa.
Nov. 4, 1880—BANCROFT, J. SELLERS....Mgr. Wm. Sellers & Co. (Incorp.), 1600 Hamilton St., and 3310 Arch St., Philadelphia, Pa.
May 21, 1884—BARNABY, CHAS. W..............Phenix I. W., Meadville, Pa.
April 7, 1880—BARNARD, GEORGE A...Steam Power Plants and Steam Htg. Apparatus, 15 Cortlandt St., Room 62, and 2023 7th Ave., N. Y. City.
May 15, 1889—BARNES, ABEL T...Consulting Eng. (for B. F. Sturtevant), Jamaica Plain, and 23 Oakdale St., Jamaica Plain, Mass.
Oct. 16, 1888—BARNES, DAVID LEONARD....Cons. Eng., 507 The Rookery, Chicago, Ill.
May 27, 1885—BARNES, PHINEHAS...P. O. Box 1021, and Jones and Laughlins, Pittsburgh, Pa.
May 26, 1886—BARNES, W. F.....Pres. W. F. & J. Barnes Co., Rockford, Ill.
Nov. 6, 1884—BARNHURST, H. R..........Sec. & Mgr. Union I. W., and 334 W. 6th St., Erie, Pa.
May 31, 1887—BARNUM, GEORGE S....Treas. "The Bigelow Co.," New Haven, Conn.
May 27, 1885—BARR, J. N.....Supt. Mo. P. C. M. & St. P. R. R., and 3028 Willis St., West Milwaukee, Wis.

* Vice-President, 1887-89. † Manager, 1887-90. ‡ Manager, 1888-91.

May 15, 1889—BARR, JOHN H....Prof. Mech. Eng'g, Univ. of Minnesota, and 1223 Fourth St., S. E., Minneapolis, Minn.
April 7, 1880—BARR, WILLIAM M....V.-P. Barr Pumping Eng. Co., Germantown Junction, Philadelphia, Pa.
Oct. 16, 1888—BARRATT, EDGAR GRANT....Prest. Exhaust Vent. Co., 64 So. Canal St., Chicago, Ill.
May 4, 1883—BARRUS, GEORGE H....Expert and Cons. Steam Eng., Room 45 Mut. Life Ins. Co. Building, 95 Milk St., Boston, Mass.
Oct. 16, 1888—BARTLETT, GEORGE B...Mech. D'tsman Union Steel Co., 3179 Ashland Ave., Chicago, Ill.
May 26, 1886—BARTOL, GEO..................Otis Steel Co., Cleveland, Ohio.
May 27, 1885—BASSETT, NORMAN C.......Mech. Eng. Otis Elev. Co., Yonkers, N. Y.
April 7, 1880—BATCHELOR, CHARLES...Orange, N. J., and 33 West 25th St., New York City.
Nov. 19, 1889—BATES, ALEX. B....Passed Asst. Eng. U. S. N., U. S.S. "Minnesota," 140 Henry St., Brooklyn, N. Y.
April 7, 1880—BAUER, CHARLES A.....Gen. Mgr. The Warder, Bushnell & Glessner Co., Springfield, Ohio.
May 15, 1889—BAUGH, SAMUEL ANDREW....Supt. Baugh Steam Forge Co., Detroit, Mich.
April 7, 1880—BAYLES, JAMES C......Spiral Weld Tube Co., 5-7 Beekman St., New York City.
Nov. 1, 1883—BEARDSLEY, ARTHUR..Prof. of Engineering and Dir. of Mech. Arts, Swarthmore College, Swarthmore, Del. Co., Pa.
May 26, 1886—BECK, MATTHIAS A..609 So. Webster St., East Saginaw, Mich.
May 15, 1889—BEEKMAN, JOHN V....Supt. Lidgerwood Mfg. Co., 383 Union St., Brooklyn, N. Y.
Nov. 29, 1887—BELCHER, AMHERST W....Supt. Repair Shops Cornell Steamboat Co., Rondout, N. Y.
Nov. 30, 1886—BELLHOUSE, R. WYNYARD..Mech. Eng'r The Warners Portland Cement Mfg. Co., Warners and Memphis, N. Y.
May 31, 1887—BELLINGRODT, M. O......Asst. Eng. Milwaukee Gas Co., Box 211, Milwaukee, Wis.
May 15, 1889—BENJAMIN, PARK....Mech. and Elect. Eng'r, 32 Park Place, New York City.
June 13, 1883—BENNETT, EDWIN H.........................Bayonne, N. J.
May 26, 1886—BENNETT, FRANK M.....Asst. Eng. U. S. N., Navy Dept., Washington, D. C.
Aug. 10, 1881—BETTS, ALFRED....Prest. Betts Machine Co., Wilmington, Del.
June 13, 1883—BETTS, WILLIAM...V.-P. Betts Machine Co., and 1211 Gilpin Ave., Wilmington, Del.
May 31, 1887—BIGELOW, FRANK L.....Secretary The Bigelow Co., and 490 Orange St., New Haven, Conn.
Nov. 2, 1882—BIGELOW, GEORGE W ....New Haven, Conn.
May 27, 1885—BILGRAM, HUGO....438 No. 12th St., cor. Noble, and 1831 Fairmount Ave., Philadelphia, Pa.
April 7, 1880—BILLINGS, CHARLES E......Prest. Billings & Spencer Co., Hartford, Conn.
Nov. 2, 1882—BINSSE, HENRY LEON........Eng. and Prop. of the Newark Machine Tool Works, Newark, N. J.

May 8, 1888—Birkinbine, John....Civ. and Min'g Eng'r, 25 N. Juniper St., and 4206 Spruce St. Philadelphia, Pa.
May 8, 1888—Bixby, William H. (*Life Member*).....U. S. A., Capt. U. S. Engr. Corps, Wilmington, N. C.
Nov. 19, 1889—Blair, Horatio P ....Steam Heating and Ventilating Eng., with E. P. Bates, 228 West Water St., Syracuse, and 88 Troup St., Rochester, N. Y.
Nov. 30, 1886—Blake, Francis C....Mgr. Penn. Lead Co., 61 Fourth Ave., Pittsburgh, Pa., and Ingram Sta., P. C. & St. L. R'y, Pa.
May 15, 1889—Blake, Percy M...........Civil Engineer, Hyde Park, Mass.
May 26, 1886—Bleloch, Geo. H.........................Springfield, Mass.
May 31, 1887—Bole, Wm. A.....Supt. Westinghouse Mach. Co.'s Works, Cor. 25th and Liberty Sts., and 4512 Howe St., E. E., Pittsburgh, Pa.
May 14, 1890—Boies, H. M........Prest. & Mgr. Boies Steel Wheel Co., & Moosic Powder Co., and 530 Clay Ave., Scranton, Pa.
Oct. 16, 1888—Bolsinger, C......................68 Rush St., Chicago, Ill.
Aug. 10, 1881—Bond, Geo. M.*..M'n'g'r Gauge Dept., Pratt & Whitney Co., and 141 Washington St., Hartford, Conn.
June 13, 1883—Bone, Wm. H.........Manager the Walker Mfg. Co., and 227 Twenty-third Av., Cleveland, Ohio.
May 21, 1884—Booraem, J. Van Vorst....Engineer-in-Chief, Havemeyer & Elder, etc., Sugar Refineries, and 204 Lincoln Place, Brooklyn, N. Y.
May 8, 1888—Booth, Thomas C....T. C. Booth, 14 Howard St., and 127 West 12th St., New York City.
May 26, 1886—Borden, Thomas J.†...... Pres. Richard Borden Mfg. Co., Fall River, Mass.
May 21, 1884—Boyd, Jas. T........Constr. Engineer Atlantic Works, and 166 Falmouth St., East Boston, Mass.
Nov. 29, 1887—Boyd, John T..... .... Gen. Man. Stearns Mfg. Co., Erie, Pa.
May 8, 1888—Bray, Charles W........Sec'y The Lloyd Booth Co., and 820 E Federal St., Youngstown, Ohio.
Aug. 10, 1881—Brady, James..Man. & Treas. Brady Mfg. Co., York and Washington Sts., and 162 Lefferts Place, Brooklyn, N. Y.
Nov. 30, 1886—Briggs, Chas. C...........Jones & Laughlins, Pittsburgh, Pa.
May 14, 1890—Bristol, W. H..... .Assist. Prof. Mathematics, Stevens Inst. Tech., Hoboken, N. J.
Nov. 19, 1889—Britton, J. W ....Pres. Britton Iron & Steel Co., 1516 Euclid Av., Cleveland, O.
May 15, 1889—Broadbent, Charles L.......Eng'r and Des. for Knowles Pump Works, 93 Liberty St., New York City.
May 21, 1884—Broadbent, Sidney.............Supt. Dickson Mfg. Co., Scranton, Pa.
May 21, 1884—Brooks, Edwin C.. Engineer Cambridge W. W. Pumping Station, Cambridge, Mass.
Nov. 11, 1885—Brooks, Morgan..Sec. and Treas. The St. Paul Gas Light Co., and 529 Holly Av., St. Paul, Minn.
May 15, 1889—Brooks, Wm. B...Chief Eng'r U. S. N., Richmond L. & M. Works, and 407 E. Main St., Richmond, Va.

* Vice-President, 1888-90. † Manager, 1888-91.

May 21, 1884—BROWN, A. G....Globe Iron Works, Kesten, Gilnow Park, Bolton, Lancashire, England.
June 13, 1883—BROWN, ALEXANDER E..V. P. & Man. The Brown Hoisting and Conveying Mach. Co., cor. Belden & Hamilton Sts., and 1151 Prospect St., Cleveland, O.
May 14, 1890—BROWN, ALEX. T....Mech. Exp't, Smith Premier Typewriter Co., 205 Shonnard St., Syracuse, N. Y.
April 7, 1880—BROWN, C. H ....C. H. Brown & Co., Fitchburg, Mass.
May 14, 1890—BROWN, CHAS. S.....Supt. Shops & Instr. Mach. Des. Rose Polytechnic Inst., Terre Haute, Ind.
May 15, 1889—BRUECK, HENRY T..Master of Mach'y, Cumb. & Penna. R.R., Mt. Savage, Md.
June 20, 1880—BRUSH, CHARLES F..Electrical Engineer, 71 Ontario Street, Cleveland, Ohio.
May 27, 1885—BULKLEY, HENRY W........Mech. Eng'r, *Times* Building, Park Row, New York City, and E. Orange, N. J.
Nov. 2, 1882—BULLOCK, MILAN C.....President M. C. Bullock Mfg. Co., Rooms 509-510 Phœnix Bldg., 13 E. Jackson St., and 1187 Washington Boul'd, Chicago, Ill.
April 7, 1880—BURDEN, JAMES A..........Pres. Burden Iron Co., Troy, N. Y.
June 13, 1883—BURGDORFF, THEO. F. (*Life Member*)....Passed Assist. Eng. U. S. N., University of Tenn., Knoxville, Tenn.
Nov. 19, 1889—BURPEE, GEO. H........Eng. & Contractor, 53 Broadway, N. Y. City, and 530 Nostrand Ave., Brooklyn.
May 21, 1884—BURR, JOHN T................Mansion House, Brooklyn, N. Y.
May 26, 1886—BUTTERWORTH, JAMES...York and Cedar Streets, Philadelphia, Pa.
Nov. 2, 1882—BYLLESBY, H. M.......Westinghouse Elec. Co., Box 1000, Pittsburgh, Pa.

May 15, 1889—CADWELL, WM. D .....Agt. Jackson Co., 21 Amory St., Nashua, N. H.
May 21, 1884—CAIRD, ROBERT.......Managing Director Caird & Co., Ltd., 5 Newark St., Greenock, Scotland.
April 19, 1882 } CALDWELL, ANDREW J.....Mech. Eng., H. R. Worthington,
May 14, 1890 } 86 Liberty St., N. Y.
Aug. 10, 1881—CAMPBELL, A. HAMILTON..Curator St. Paul's School, P. O. Box 370, Concord N. H.
Nov. 11, 1885 } CAMPBELL, ANDREW C.....Chief D'ftsman Farrell F. & M.
Nov. 19, 1889 } Co., and 26 N. Willow St., Waterbury, Conn.
Nov. 1, 1883—CAMPBELL, GEO. W..Constr. Eng., 195 Broad St., Newark, N.J.
May 14, 1890—CANNING, WM. PITT....Chf. Dftsman. Mach. Dept. Lowell Mach. Shops, and 6 Belmont St., Lowell, Mass.
June 13, 1883—CAPEN, THOMAS W....Chief D'tsman. Fraser & Chalmers, Fulton & Union Sts., and 227 Warren Ave., Chicago, Ill.
Nov. 1, 1883—CARPENTER, R. C..Assoc. Prof. of Exp. Mechanics, Sibley College, Cornell Univ., Ithaca, N. Y.
Nov. 19, 1889—CARSE, DAVID BRADLEY....Gen. M'g'r. Greenlee Bros. & Co., 225-235 W. 12th St. and 145 Ashland Boulev., Chicago, Ill.
May 21, 1884—CARR, C. A..Ass't Eng'r U. S. N., Navy Dept., Washington, D. C.

May 8, 1888—Carroll, Lafayette D...Mech. Eng., care Carroll & Carroll, Att'ys-at-Law, Birmingham, Ala.

May 14, 1890—Carter, Vaux ...Chf. Dept. Mech. Drawing & Mach'y, 34-36 Stuyvesant St., N. Y., and 181 Schermerhorn St., Brooklyn, N. Y.

May 21, 1884—Cartwright, Robert....C. & M. Eng., 1 Hawthorne St., Rochester, N. Y.

May 15, 1889—Cary, Albert A..Mech. Eng. Cary & Moen Co., 234 W. 29th St., N. Y. City.

May 26, 1886—Cavanagh, Joseph..........Eng. The Link Belt Eng'g Co., Nicetown, and 18th and Vine Sts., Philadelphia, Pa.

May 21, 1884—Chamberlin, Franklin L....Sec'y The Variety Iron Works Co., 51-73 Scranton Ave., and 909 Case Ave., Cleveland, O.

Aug. 10, 1881—Chapman, Luke............M. M. The Collins Co., Box 34, Collinsville, Conn.

Nov. 29, 1887—Charnock, John Milton..Cons. and Htg. Engr., 16 Dey St., New York City.

Nov. 1, 1883—Cheney, Walter L....Sec'y Meriden Mach. Tool Co., Box 702, and 126 Pleasant St., Meriden, Conn., also 55 Huyshope Ave., Hartford, Conn.

Nov. 19, 1889—Christiansen, Alfred...Mech. Eng. & Supt. Gun Works, U. S. Arsenal, and 1231 Fifth Av., W. Troy, N. Y.

Nov. 4, 1880—Christensen, August C....Supt. Snow S. P. Wks., C. & I. Exch., and 77 West Utica St., Buffalo, N. Y.

Nov. 11, 1885—Christie, James ...... Pencoyd Iron Works, Pencoyd, and Wissahickon, Phila., Pa.

May 15, 1889—Christie, W. Wallace.......Asst. Eng'r, Ramapo, I. W., Hillburn, N. Y., and 203 Summer St., Paterson, N. J.

April 7, 1880—Church, Wm. Lee*.....Westinghouse, Church, Kerr & Co., 620 Atlantic Ave., Boston, and Newton Centre, Mass.

Nov. 4, 1880—Churchill, Thomas L...Insurance Inspector, 71 Kilby St., Room 30, Boston, and 17 Fifth St., Chelsea, Mass.

May 14, 1890—Cité, Joseph D......Des. & Cons. Eng., Fishkill Landing Mach. Co., Fishkill-on-Hudson, N. Y.

May 8, 1888—Clark, Walter L ....N. Y. Mgr. Niles Tool Works, 98 Liberty St., New York City.

May 14, 1890—Clark, Walton......Assist. Gen. Supt. The United Gas Imp't Co., 813 Drexel Bldg., Phila., Pa.

May 26, 1886—Clarke, Alfred..Vice-Prest. & Genl. Mgr. The Eastern Elect. Light & Storage Battery Co., Lowell, Mass.

Nov. 2, 1882—Clarke, Charles L.........Cons. Eng. & Patent Expt., 55 Liberty St., N. Y., and 40 Walnut St., East Orange, N. J.

June 13, 1883—Clarke, David.............M. M., L. V. R. R., Hazelton, Pa.

May 21, 1884—Clarke, Samuel J.....Supt. Eng'r. Prov. and Stonington S. S. Co., New Pier 36, N. R., and 205 W. 43d St., New York City.

May 14, 1890—Clarke, T. C.............Cons. Eng., 1 Broadway, N. Y. City.

May 14, 1890—Clawson, Linus P..........Sec'y Black & Clawson Co., Hamilton, Ohio.

Oct. 16, 1888—Clay, John Ridgway......Hydr. Eng. and D'tsman, Smith & Vaile Co., Room 6, 2 East Third St., Dayton, O.

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* Manager, 1884-87.

May 26, 1886—CLEMENTS, WM. L...........Industrial Works, Bay City, Mich.
June 20, 1880—CLOUD, JOHN W...........Mech. Eng., 31 White Building, and 996 Main St., Buffalo, N. Y.
Aug. 10, 1881—COES, Z. B....................Niles Tool Works, Hamilton, O.
April 7, 1880—COGGIN, FREDERIC G.*...Supt. Calumet & Hecla Stamp Mills, Lake Linden, L. S., Mich.
April 7, 1880—COGSWELL, WM. B.†..Gen. Man. Solvay Process Co., 25 White Building, Syracuse, N. Y.
May 8, 1888—COLE, FRANCIS J.Chief D'tsman Mech. Dept. B. & O. R. R., Mont Clare, Baltimore, Md.
Nov. 1, 1883—COLE, J. WENDELL..Detroit Emery Wheel Co., P. O. Box 84, and 1217 East Rich St., Columbus, Ohio.
May 8, 1888—COLEMAN, ISAIAH B....Foundry and Mach. Shop, State and Church Sts., and 353 West 7th St., Elmira, N. Y.
May 14, 1890—COLEMAN, JOHN A......Mech. Eng. & Commissioner Public Works, Providence, R. I.
May 8, 1888—COLEMAN, WM. H.............I. & M. Wolff and Coleman, 261 Dearborn St., Chicago, Ill.
May 26, 1886—COLLIER, R. B. Supt. Columbus Mach. Co., 139 W. Broad St., Columbus, Ohio.
May 26, 1886—COLLINS, CHAS. M.............. ...Supt. Studebaker Bros,, and 117 E. Tutt St., South Bend, Ind.
April 7, 1880—COLLINS, C. C.................10 Rowland St., Newark, N. J.
April 7, 1880—COLWELL, AUGUSTUS W..74 Cortlandt St., and 365 W. 27th St., New York City.
Nov. 5, 1880—COMLY, GEORGE N......Cons. Mech. Eng., Edgemoor Iron Works, Edgemoor, Del.
June 13, 1883} CONANT, THOMAS P......General Supt. Constr'n, United
Nov. 19, 1889} Edison Mfg. Co., 65 Fifth Ave., N. Y. City.
May 15, 1889—COOK, A. S....Manuf. of Machinery, Colt's West Armory Hartford, Conn
Nov. 30, 1886—COOK, FREDERIC..Consulting and Mech. Eng., 57 Carondelet St., Box 2524, New Orleans, La.
May 21, 1884—COOLEY, M. E..Prof. Mech. Eng. University of Michigan, and 32 Packard St., Ann Arbor, Mich.
April 7, 1880—COON, J. S..........Prof. Eng'g School of Tech., Atlanta, Ga.
April 7, 1880—COOPER, JOHN H......Mech. Eng., Southwark Foundry and Mach. Co., 4724 Springfield Ave., Philadelphia, Pa.
April 7, 1880—COPELAND, CHARLES W.‡..24 Park Place, New York City, and 151 Columbia Hts., Brooklyn, N. Y.
April 7, 1880—COPELAND, GEORGE M.........24 Park Place, New York City.
May 21, 1884—CORBETT, CHAS. H........Continental Iron Works, and 428 Lafayette Ave., Brooklyn, N. Y.
Nov. 30, 1886—CORLISS, WILLIAM..........Pres. Corliss Safe Mfg. Co., Providence, R. I.
May 14, 1890—CORRY, WM..........Supt. Hall's Safe & Lock Co., Cinn., O.
Oct. 16, 1888—CORNELIUS, HENRY ROBERT.....Mech. Eng. Southwark F. & M. Co., 430 Washington ave., and Highland Ave., Chestnut Hill, Philadelphia, Pa.

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* Manager, 1885-88. † Manager, 1880-82.
‡ Treasurer, Dec. 2, 1881, to Nov. 7, 1884; Vice-President Nov. 7, 1884, to Dec. 2, 1886.

April 7, 1880—COTTER, JOHN.................................Mt. Holly, N. J.
Nov. 30, 1886—COTTRELL, CALBERT B..Printing Presses, etc., 8 Spruce St., New York City.
Aug. 10, 1881—COWLES, WILLIAM..Const. Eng. & Naval Arch't, 45 Broadway, New York City, and 434 Eighth St., Brooklyn.
Aug. 10, 1881—COX, J. D., JR....Gen. Man. Cleveland Twist Drill Co., cor. Lake and Kirtland Sts., and 468 Euclid Ave., Cleveland, Ohio
April 7, 1880—COXE, ECKLEY B.*.................Drifton, Luzerne Co., Pa.
May 15, 1889—CRAMP, ANDREW D..Naval Architect, Wm. Cramp & Sons, Beach and Norris Sts., Philadelphia, Pa.
Oct. 16, 1888—CRAMP, EDWIN S......Supt. Eng. Wm. Cramp & Sons, S. E. & B. Co., Beach and Ball Sts., Philadelphia, Pa.
May 31, 1887—CRANE, WILLIAM EDWARD..Chief Eng. Benedict & Burnham Mfg. Co., 37 Baldwin Hill, Waterbury, Conn.
May 27, 1885—CREELMAN, WM. J....Engineer Woodbury Engine Co., and 119 Ambrose St., Rochester, N. Y.
Nov. 11, 1885—CREMER, JAMES M ...H. R. Worthington Hydraulic Works, So. Brooklyn, N. Y.
May 14, 1890—CROCKER, JOHN B...Supt. Niles Tool Works, and 1028 Vine St., Hamilton, Ohio.
Aug. 10, 1881—CROSBY, GEORGE H....Crosby Steam Gauge and Valve Co., 95 Oliver St., Boston, Mass., and Elm Farm, Albion, Me.
May 27, 1885—CROUTHERS, JAMES A....Coal and Iron Exchange, Room 6, Cortlandt and Church Sts., New York City, and Astoria, L. I., N. Y.
Nov. 29, 1887—CROWELL, LUTHER C..........R. Hoe & Co., 504 Grand St., New York City, and 174 Hoop St., Brooklyn.
May 15, 1889 }
May 14, 1890 } CRUIKSHANK, BARTON..Pres. and Assist. Man. Brady Mfg. Co. York and Washington Sts., and 206 So. Oxford St., Brooklyn, N. Y.
Nov. 30, 1886—CULLEN, JAMES K.....................96 Lake St., Chicago, Ill.
May 21, 1884—CULLINGWORTH, GEO. R.....49 West 93d St., New York City.
June 13, 1883—CUMMER, F. D....Cummer Engine Co., 23 Hawthorne Ave., Cleveland, Ohio.
May 21, 1884—CUMMINGS, A. G..Manager Excelsior Elect. Co., Walnut St., and P. R. R., Harrisburg, Pa.
Aug. 10, 1881—CURTIS, GRAM....Cons. Mech. Eng., 612 and 613 Lewis Block, Pittsburgh, Pa.
April 7, 1880—CUSHING, G. W..Supt. M. P. and Mach'y U. P. Ry., Omaha, Neb.

May 27, 1885—DAGRON, JAMES G..........Eng'r of Bridges, B. & O. Ry., Baltimore, Md.
Oct. 16, 1888—DALLETT, W. P....Wm. Sellers & Co., Incorporated, 16th and Hamilton Sts., and 3206 Summer St., Phila., Pa.
Oct. 7, 1881—DANFORTH, ALBERT W ....Engineer and Supt. Shanghai Cotton Cloth Mills Co., Shanghai, China, also 245 Bridge St. Lowell, Mass.
May 26, 1886—DANIELS, FRED. H....Washburn & Moen Mfg. Co., and 130 Lincoln St., Worcester, Mass.
April 7, 1880—DARLEY, E. C.. ....P. A. U.

* Vice-President, 1880-81.

May 14, 1890—DARLINGTON, F. G...Supt. I. & S. Div. C. St. L. & P. R'y., Indianapolis, Ind.

May 14, 1890—DASHIELL, W. W..Gen. Mgr. Sloss I. & S. Co., and 17th St. near 8th Ave., Birmingham, Ala.

Nov. 30, 1886—DAVIDSON, MARSHALL T....Pumping Mach'y, 43 Keap St., Brooklyn, and 108 St. James Place, B'klyn, N. Y.

June 20, 1880—DAVIES, R. H....M. M., Phœnix Iron Works, Phœnixville, Pa.

May 14, 1890—DAVIS, CHAS. H....Cons. Eng., 120 Broadway, and 576 Lexington Ave., N. Y.

May 14, 1890—DAVIS, CHESTER B....Cons. and Hydraulic Eng., 549 "The Rookery," and 3762 Lake Ave., Chicago, Ill.

April 7, 1880—DAVIS, DAVID P....Engineer N. Y. Safety Steam Power Co., 30 Cortlandt St., New York City, and Allendale, Bergen Co., N. J.

May 15, 1889—DAVIS, D. W.............Supt. Buckeye Eng. Co., and 258 Lincoln Ave., Salem, O.

Nov. 4, 1881—DAVIS, E. F. C ......Gen. Mgr. Richmond L. & M. Wks., Richmond, Va.

May 27, 1885—DAVIS, ISAAC H. ...Westinghouse, Church, Kerr & Co., 17 Cortlandt St., N. Y., and Dorchester, Mass.

Nov. 4, 1880—DAVIS, JOSEPH P......Phenix Constrn. Co., 115 W. 38th St., Telephone Bldg., New York City.

June 13, 1883—DEAN, FRANCIS W...........Const. and Expert M. E., 27 School St., Boston, Mass.

April 7, 1880—DEANE, CHARLES P....Treas. Deane Steam Pump Works, Holyoke, Mass.

May 27, 1885—DEBES, J. C.....M. E. and Supt. C. & G. Cooper & Co., Mt. Vernon, Ohio.

May 14, 1890—DELANEY, ALEXANDER..Mech. Eng. Richmond L. & M. Wks., Richmond, Va.

Nov. 11, 1885 }
Nov. 23, 1887 } DENT, EDWARD LINTHICUM....Dent's Iron Wks., Water and 33d St., and 3101 N St., Washington, D. C.

May 4, 1881—DENTON, JAMES E*.. Prof. Exper'l Mech. Stevens Inst. Tech., Hoboken, and 83 Sipp Ave., J. C. Hts., N. J.

May 14, 1890—DERBYSHIRE, WM. H....Mech. Eng. with Bement, Miles & Co., 21st and Callowhill Sts., Phila., Pa.

April 7, 1880—DE SCHWEINITZ, P. B....Steel Works, Colorado Coal and Iron Co., South Pueblo, Col.

Nov. 29, 1887—DE VALIN, CHARLES E..............Chf. Eng'r U. S. N., 1308 New Hampshire Ave., Washington, D. C.

Nov. 30, 1886—DICEY, ELMER C......Constr. Eng., 68-70 W. Monroe St., and 1225 Lexington St., Chicago, Ill.

May 21, 1884—DICKEY, WM. D....Supt. Albany Street Iron Works, 126 Washington St., New York City.

May 26, 1886—DINGEE, W. W... Mech. Eng. J. I. Case, T. M. Co., and 1124 Main St., Racine, Wis.

May 21, 1884—DIXON, CHAS. A....Supt. Newburgh Steam Eng. Works, Newburgh, N. Y.

May 31, 1887—DIXON, GEORGE EDWARD. ..Mech. Eng'r The John Davis Co., 69-79 Michigan St., Chicago, Ill.

* Manager, 1889-92.

Nov. 6, 1884—DIXON, ROBERT M....Eng. Safety Car Htg. and Lighting Co., 160 Broadway, New York City, and 21 Walnut St., East Orange, N. J.

May 27, 1885—DOANE, WM. H....Pres't J. A. Fay & Co., 267-285 Front St., Cincinnati, Ohio.

May 4, 1881—DOBSON, W. J. M..Mech. Eng., 96 Albany Ave., Brooklyn, N.Y.

May 15, 1889—DOCK, HERMAN....Mech. Eng. Schlicter Jute Cordage Co., 21 N. Front St., and 904 N. Broad St., Philadelphia, Pa.

May 31, 1887—DODDS, ELIHU......D'tsman, Chandler and Taylor Co., 164 Broadway and 956 No. Delaware St., Indianapolis, Ind.

May 21, 1884—DODGE, JAMES M....Chief Eng'r Link Belt Eng'g Co., Nicetown, and 2027 Arch St., Philadelphia, Pa.

Nov. 1, 1883—DONOVAN, WM. F......Manager Yale & Towne Mfg. Co., 152 & 154 Wabash Ave., Chicago, Ill.

May 15, 1889—DORAN, WILLIAM S.....Eng'r with H. R. Worthington, 145 Broadway, and 115 Waverly Place, New York City.

May 21, 1884—DOUGLASS, WM. M........Gen. Supt., Iowa Barb Wire Co., Allentown, Pa.

May 26, 1886—DOWNE, HENRY S..................108 Lake St., Chicago, Ill.

May 15, 1889—DRAPER, T. W. MORGAN....Chf. Eng'r, Atl. and Danville Ry., Portsmouth, and Atlantic Hotel, Norfolk, Va.

Oct. 16, 1888—DREWETT, WM. A.....Supt. Davidson Stm. Pump Co., 41-51 Keap St., and 130 Rutledge St., Brooklyn, N. Y.

May 15, 1889—DROWN, FREDERICK EUGENE....Architect, Mech. and Cotton Mill Eng'r, 177 Main St., and 22 Spring St., Pawtaucket, R. I.

May 15, 1889—DRUMMOND, D. D., Eng'r Scoville I. W., 250-254 So. Clinton St., and 372 Claremont Ave., Chicago, Ill.

Aug. 10, 1881—DRUMMOND, W. W............Pres. Drummond Mfg. Co., Louisville, Ky.

May 14, 1890—DRYSDALE, W. A........Cons. Eng., 1531 Arch St., Phila., Pa.

Aug. 10, 1881—DUBOIS, A. JAY (*Life Member*)...Prof. Civ. Eng. S.S.S., Yale Univ., New Haven, Conn.

June 13, 1883—DUDLEY, CHARLES B.......Chemist, Penn. R. R. Co., and 1219 Twelfth Ave., Altoona, Pa.

Nov. 1, 1883—DUNCAN, JOHN.........Asst. Supt. Calumet & Hecla Mine, Calumet, Houghton Co., Mich.

Nov. 1, 1883—DURAND, WM. F.......Prof. Mech. Eng., Michigan Agricul. Col. (Lansing), Ingham Co., Mich.

April 7, 1880—DURFEE, W. F.*..General Manager, Penn. Diamond Drill Co., Birdsboro, Berks Co., Pa.

May 31, 1887—DUTTON, C. SEYMOUR....Gen. Agent and Consulting Eng. Hamilton Works, Wm. Tod & Co., and 656 Bryson St., Youngstown, Ohio.

Nov. 6, 1884—DU VILLARD, HENRY A....Mech. Eng'r, Granger Foundry and Mach. Co., Gaspee and Francis Sts., Prov., R. I.

May 8, 1888—EASBY, FRANCIS H....Mech. Eng'r Betts Machine Co., Wilmington, Del.

June 13, 1883—EASTWICK, GEORGE S....Louisiana Sugar Refining Co., New Orleans, La.

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* Manager, 1883-86.

April 19, 1882—ECKART, W. R.*....P. O. Box 1844, and 217 Samson St., Rooms 2 and 3, San Francisco, Cal.
April 7, 1880—EDISON, THOS. A. (*Life Member*)................Orange, N. J.
June 13, 1883—EDSON, JARVIS B..Prop. and Mfg'r Pressure Record'g Gauges and Exact Apparatus, 87 Liberty St., New York, and 812 Union St., Brooklyn, N. Y.
April 7, 1880—EGLESTON, THOS...Prof. Metallurgy, School of Mines, Columbia College, and 35 Washington Square, West, New York City.
May 14, 1890—EHBETS, C. J.....Mech. Eng., Colt's Pat. Fire-Arms Co., and 68 Washington St., Hartford, Conn.
May 14, 1890—EHLERS, PETER.....Mfr. and constr , 170 B'way, Albany, and Clinton Heights, N. Y.
Nov. 1, 1883—ELMES, CHAS. F....Eng., Founder and Machinist, Fulton and Jefferson Sts., Chicago, Ill.
April 7, 1880—ELY, THEO. N.†...Gen. Supt. Motive Power, Penn. R. R., Altoona, Pa.
Nov. 19, 1889—ELLIOTT, W. E......Chf. Eng. Goodrich Transportation Co., Manitowoc, Wis., also Chicago, Ill.
April 7, 1880—EMERY, ALBERT H...............Civil Eng., Stamford, Conn.
April 7, 1880—EMERY, CHARLES E.‡...Consulting Eng'r, 22 Cortlandt St., New York City, and 370 Greene Ave., Brooklyn, N. Y.
May 31, 1887—ENGEL, LOUIS G....Brooklyn Sug. Ref. Co., and 238 Clermont Ave., Brooklyn, N. Y.
Nov. 6, 1884—EWART, WILLIAM D ...Consult. Eng. Ewart Mfg. Co., 11 So. Jefferson St., Chicago, Ill., and Pres. Link Belt Eng'g Co., Nicetown, Philadelphia, Pa.
May 31, 1887—EWEN, JOHN MEIGGS....Architectural Engineer, Room 1142, The Rookery, Chicago, Ill.
April 7, 1880—EWER, ROLAND G....Supt. Penn. Salt Mfg. Co., Natrona, Allegheny Co., Pa.

April 7, 1880—FABER DU FAUR, A.. Room 56 Vanderbilt Bldg., 132 Nassau St., New York City, and 22 Nichols St., Newark, N. J.
May 15, 1889—FAIRBAIRN, W. U.......Chief Inspector Hartford S. B. Insp. and Ins. Co., Hyde Park, Mass.
May 26, 1886—FALKENAU, ARTHUR....Mech. Eng. and Macht., 11th St. and Ridge Ave., and 3214 Spencer Terrace, Phila., Pa.
Nov. 2, 1882—FARMER, MOSES G. ....Electrical Engineer, Mass., and Newport, R. I.
June 13, 1883—FAWCETT, EZRA..Eng'r and Prop'r Alliance Industrial Works, Ely Ave., Alliance, Ohio.
June 13, 1884—FAY, RIMMON C.......Supt. Remington Arms Co., Ilion, N. Y.
May 27, 1885—FELTON, EDGAR CONWAY...Supt., Penn. Steel Co., Steelton, Pa.
Oct. 16, 1888—FICKINGER, P. J..........Chf. Eng. Carnegie, Phipps & Co., Beaver Falls, Pa.
May 15, 1889—FIELD, CORNELIUS J.......Field Eng'g Co., 15 Cortlandt St., New York City.
May 26, 1886—FINGAL, CHAS. A..........Supt. A. Sandstrom & Co., cor. Root and La Salle Sts., Chicago, Ill.

* Vice-President, 1883-85. † Manager, 1880-82; Vice-President, 1881-82.
‡ Vice-President, 1881-83.

April 7, 1880—FIRMSTONE, FRANK....................................Easton, Pa.
May 21, 1884—FISHER, CHAS. H....................................Canton, Mass.
May 26, 1886—FITT, JAMES........Supt. Shipman Eng. Co., Rochester, N. Y.
Nov. 29, 1887—FLADD, FREDERICK C...N. Y. Mgr. Stiles & Parker Press Co., 203-207 Centre St., and 203 East 69th St., N. Y. City.
May 15, 1889—FLATHER, JOHN J....Instr. Mech. Eng'g, Lehigh University, and 140 Market St., Bethlehem, Pa.
Nov. 19, 1889—FLETCHER, ANDREW..Prest. and Treas. W. & A. Fletcher Co., 266 West St., and 157 W. 73d St., New York City.
Nov. 19, 1889—FLETCHER, W. H.....Chf. Dftsman W. & A. Fletcher Co., 266 West St., New York City.
Nov. 30, 1886—FOLGER, WILLIAM MAYHEW....Comm'r U. S. N., Head of Ord. Dept., Navy Yard, Washington, D. C.
May 26, 1886—FORBES, WM. DUNDERDALE....Mech. Eng'r, Morristown, N. J.
May 24, 1884—FORD, JOHN D....Engineer Corps, U. S. N., 1522 W. Lauvale St., Baltimore, Md.
April 7, 1880—FORNEY, MATTHIAS N...Ed. *R. R. and Engineering Journal*, 45 Broadway, and 431 Fifth Ave., New York City.
May 4, 1881—FORSYTH, ROBERT..........Chf. Eng. Illinois Steel Co., 1035 "Rookery," Chicago, Ill
Nov. 1, 1883—FORSYTH, WILLIAM *..Mech. Eng'r C. B. & Q. R. R., Aurora, Ill.
May 26, 1886—FOSTER, C. H... Fraser & Chalmers, Fulton and Union Sts., Chicago, Ill.
May 8, 1888—FOSTER, WILLIAM A... Supt. Mo. P. and Mach'y Fall Brook Coal Co., and 130 East 3d St., Corning, N. Y.
May 26, 1886—FOWLER, GEO. L.........170 Broadway, New York City, N. Y.
May 21, 1884—FOWLER, JOHN..........Eagle Brass Works, 225 Eighth St., Louisville, Ky.
Oct. 16, 1888—FOWLER, PERCIVAL.....Min'g and Cons. Eng., P. Fowler and Power, 16 St. Helen's Place, London, E. C., England.
Nov. 6, 1884—FRANCIS, HARRY C........Steam Eng'g Co., 704 Arch St., Philadelphia, Pa.
May 26, 1886—FRANCIS, JAS......Agent and Chief Eng'r, Prop's Locks and Canals, Merrimac River, 22 Broadway, Lowell, Mass.
May 21, 1884—FRANCIS, W. H....Sec'y Kensington Eng. Wks. (Ltd.), Beach and Vienna Sts., and 1732 Master St., Phila., Pa.
May 26, 1886—FRASER, DAVID R.....Fraser & Chalmers, Fulton and Union Streets, Chicago, Ill.
May 26, 1886—FRASER, NORMAN D....Fraser & Chalmers, Fulton and Union Streets, Chicago, Ill.
May 21, 1884—FREELAND, FRANCIS T..Cons. Mining & Mech. Eng., Box 23, Leadville, Col.
May 31, 1887—FREEMAN, JOHN R......Eng'r & Spec. Insp. Assoc. Factory Mut. Ins. Co., 31 Milk St., Boston, and Main St., Winchester, Mass.
May 15, 1889—FRENCH, E. C...The Rathbun Co., and Supt. Deseronto Chemical W'ks and Gas Co., Deseronto, Ontario, Canada.
May 27, 1885—FRICK, ABRAHAM O..........Frick Mfg. Co., Waynesboro, Pa.
May 4, 1881—FRITZ, JOHN *....Chief Eng. and Gen. Supt. Bethlehem Iron Works, Bethlehem, Pa.

* Manager, 1888-91.

April 7, 1880—Fuller, Levi K..Vice-Prest. and Supt., Estey Organ Works, Brattleboro, Vt.

Nov. 29, 1887—Gage, Howard, Asst. Eng. U. S. N.....Bu. Steam Eng'g, Washington, D. C.
Nov. 30, 1886—Gale, Horace B..........Prof. Dyn. Eng'g, Washington University, and 3012 Lucas Ave., St. Louis, Mo.
April 7, 1880—Galloupe, Francis E.. Mech. Eng., 30 Kilby Street, Boston, and Cliff Ave., Winthrop Highlands, Mass.
May 8, 1888—Gantt, Henry L....Supt. Casting Dept. Midvale Steel Wks., Nicetown and 7 E. Penn St., Gmtn., Philadelphia, Pa.
May 8, 1888—Garrett, William................. Joliet Steel Co., Joliet, Ill.
May 21, 1884—Gaunt, Thomas..Gaunt & Martinez, Consult. & Contr. Eng'r, & Archit. Amer. Filtering Press Co., 115 B'way, N. Y. City.
May 31, 1887—Geoghegan, Stephen J.Steam Heating Eng'r, 116 Wooster St., New York City.
May 21, 1884—Geer, James H...Asst. Eng. Cambria S. & I. Co., and 58 Somerset St., Johnstown, Pa.
May 14, 1890—Gibbs, Geo..Mech. Eng., C., M. & St. P. RR., Milwaukee, Wis.
May 21, 1884—Giddings, C. M.....President Sioux City F'dry, Sioux City, Ia.
April 7, 1880—Gill, John L., Jr....503 Woodland Terrace, Philadelphia, Pa.
May 8, 1888—Gillis, H. A.....Gen. Fmn. Office M. M., N. Y., L. E. & W. RR., Elmira, N. Y.
May 15, 1889—Gilmore, Robert J...M'g'r The Allen Fire Dept. Supply Co., Providence, R. I.
May 26, 1886—Gobeillf, Jos. Leon..........26 York Street, Cleveland, Ohio.
May 21, 1884—Gold, Samuel F.....Sec Gold Car Heating Co., Bridge Store No. 6, N. Y. City, and Englewood, N. J.
June 13, 1883—Good, Wm. E........Supt. Southwark Foundry & Mach. Co., 430 Washington Av., and 3800 Locust St., Philadelphia, Pa.
Nov. 30, 1886—Goodale, A. M.........Agt. Boston Mfg. Co., Waltham, Mass.
Nov. 30, 1886—Goodfellow, Geo.........M. M. Penn. Steel Co., Steelton, Pa.
April 7, 1880—Gordon, Alex.....Vice-P. and Gen. Man., Niles Tool Works, Hamilton, Ohio.
June 20, 1880—Gordon, Fred. W...Gordon, Stroebel & Laureau, Mifflin and Meadow Sts., Phila., Pa.
May 26, 1886—Goss, W. F. M...Prof. Prac. Mech., Purdue Univ., Lafayette, Ind., also 49 Rutland St., Boston, Mass.
April 7, 1880—Goubert, Auguste A...Treas. Goubert M'f'g Co., 32 Cortlandt St., New York City, and 735 Quincy St., Brooklyn, N. Y.
May 31, 1887—Gould, W. V........Supt. C. B. Rogers & Co., Norwich, Conn.
May 26, 1886—Gowing, E. H........Civil and Mech. Eng'r, 70 Kilby Street, Boston, Mass.
May 21, 1884—Graham, J. S.............J. S. Graham & Co., Rochester, N. Y.
April 7, 1880—Grant, John J.... .....Supt. Simonds Rolling Machine Co., Fitchburg, Mass.
May 14, 1890—Graves, Erwin.....Chf. Eng., Camden Iron Works, and 519 Linden St., Camden, N. J.
May 21, 1884—Gray, G. A.......... .....481 Sycamore St., Cincinnati, Ohio.

* Vice-President, 1882-84.

May 14, 1890—GRAY, THOMAS...Prof. Dynamic Engineering, Rose Polytechnic Inst., and 318 W. 7th St., Terre Haute, Ind.
April 19, 1882 } GREENE, ISAAC CHASE....Green & Prunty, Hope Brass Works,
Nov. 30, 1886 } Baltimore, Md.
May 15, 1889—GREENE, STEPHEN..Lockwood, Greene & Co., Mill Archts. and Eng'rs, Rialto Bldg., 131 Devonshire St., Boston, and Newton Center, Mass.
May 14, 1890—GREEN, SAMUEL M..... .. Mech. Supt., Merrick Thread Co., Holyoke, Mass.
May 21, 1884—GREENWOOD, J. H.....Mech. and Hydr. Eng'r, 146 Water St., and 120 Clinton St., Cleveland, Ohio.
Oct. 16, 1888—GREGG, JOHN H.........Chf. D'tsman Link Belt Mach. Co., 23 S. Jefferson St., and 355 W. Adams St., Chicago, Ill.
May 14, 1890—GREGORY, WM..Mfr. Brass and Iron Fittings for Steam Engine Bldrs., cor. Cannon and Stanton Sts., N. Y. City.
May 15, 1889—GRIFFIN, EUGENE.....Capt. Corps Eng'rs U. S. A., Gen. Mgr. Ry. Dept. Thomson Houston Elect. Co., 620 Atlantic Ave., and 339 Beacon St., Boston, Mass.
May 14, 1890—GRIMM, PAUL H..............Chf. Eng., Nat'l Starch Co., Glen Cove, L. I., N. Y.
Nov. 6, 1884—GRINNELL, FREDERICK*........Pres. Providence Steam and Gas Pipe Co., Providence, R. I.
Nov. 19, 1889—GRIST, B. W.....Gen'l Supt. and Eng. Penn'a Iron Works, Fiftieth St. and Merion Ave. and 4712 Girard Ave., Phila., Pa.
May 27, 1885—GRISWOLD, FRANK L....Cons. Eng., 443 Calle Peru, Buenos Ayres, Argent. Rep., S. A.

April 7, 1880—HAGUE, CHARLES A........Mech. Eng'r, 86 Liberty St., H. R. Worthington, New York City and Hackensack, N. J.
Aug. 10, 1881—HALL, ALBERT FRANCIS........George F. Blake Mfg. Co., 3rd St., E. Cambridge, and 3 Cordis St., Charlestown, Mass.
Nov. 19, 1889—HALL, WILLIS E..Asst. M. M. P. R. R. and Box 505, Altoona, Pa.
Nov. 2, 1882—HALSEY, F. A........................Sherbrooke, Quebec, Can.
May 27, 1885—HALSEY, JAS. T...............Park House, Morristown, N. J.
May 21, 1884—HAMMER, ALFRED E......Supt. Mall. Iron Fittings Co., Branford, Conn.
May 8, 1888—HAMMETT, HIRAM G.......Mgr. Estate F. W. Richardson, 464 Eighth St., Troy, N. Y.
Nov. 6, 1884—HAMMOND GEO. W..Fiber Manufacturer, Yarmouthville, Me., and Hotel Hamilton, Boston, Mass.
Oct. 16, 1888—HAND, FRANK LUDLAM..Gen. Supt. Phila. Bureau of Water, Juniper and Filbert Sts., and 1948 N. 18th St., Philadelphia, Pa.
Nov. 2, 1882—HAND, S. ASHTON........Gen. Supt. Pascal Iron Works, Morris Tasker & Co., Icpd., 1601 S. Fifth St., and 1620 S. Broad St., Philadelphia, Pa.
May 15, 1889—HANDREN, JOHN W......Handren & Robins, Steam Eng'rs and Dry Docks, 126 Washington St., New York City.
Nov. 30, 1886—HANSON, AUGUSTUS......Cons. Eng., Leonard & Izard Co., Rialto Bldg., Chicago, Ill., and Box 83, Maplewood, Ill.

* Manager, 1887-90.

May 31, 1887—HARDIE, ROBERT....Asst. Mech. Eng., De La Vergne, Refr. Mach. Co., foot of E. 138th St., and 1007 E. 141st St., New York City.
May 21, 1884—HARMON, ORVILLE S..Supt. of Constr. for P. Lorillard & Co., Jersey City, N. J., and 318 Monroe St., Brooklyn, N. Y.
May 14, 1890—HARRIS, JOHN H..Vice-Prest. and Gen. Man. Worthington Pumping Engine Co., 145 Broadway, N.Y. City.
April 19, 1882—HARTMAN, JOHN M.....Mech. Eng., 1235 N. Front Street, Philadelphia, Pa.
Nov. 11, 1885—HARTSHORNE, WILLIAM D.... Supt. Worsted Department, Arlington Mills, and 500 Broadway, Lawrence, Mass.
May 21, 1884—HAWKINS, JOHN T.* Pres. Campbell Printing Press & Mfg. Co., Taunton, Mass.
May 27, 1885—HAYES, GEORGE......Hayes Skylights, 71 Eighth Avenue, New York City.
May 15, 1889—HAYWARD, FRED. H ...Williams & Potter, 15 Cortlandt St., New York City.
April 7, 1880—HAYWARD, H. S....Supt. Motive Power, United Railroads of N. J. Div. P. R. R., Jersey City, N. J.
Nov. 2, 1882—HAZARD, VINCENT G.....D'tsman, The Pusey & Jones Co., and 602 West St., Wilmington, Del.
Nov. 19, 1889—HEGGEM, CHARLES O....Supt. Eng. Dept. Russell & Co. and 215 E. South St., Massillon, O.
April 7, 1880—HEMENWAY, F. F....Ed. *American Machinist*, 96 Fulton St., New York City, and 109 Palisade Ave., J. C., N. J.
May 21, 1884—HEMPHILL, JAMES.....Mackintosh, Hemphill & Co. (Ltd.), foot 12th St., Pittsburgh, Pa.
May 27, 1885—HENDERSON, ALEXANDER, Chf. Eng. U. S. N., Navy Yard, Boston, Mass.
June 13, 1883—HENNEY, J. B......................93 Bird St., Boston, Mass.
May 21, 1884—HENNEY, JOHN, JR....Gen. Supt., Mo. P., N. Y., N. H. & H. R. R., and 276 Orange St., New Haven, Conn.
Nov. 4, 1880—HENNING, GUSTAVUS C......Box 22, Johnstown, Pa., and Consult'g Eng'r, 18 Cedar St., New York City.
May 31, 1887—HENRY, WILLIAM THOMAS......C. and Mech. Engineer, Fall River, Mass.
May 4, 1881—HENTHORN, JOHN T..Remington & Henthorn, Mech. Eng'rs, P. O. Box 1271, and 146 Westminster St., Room 21, Providence, R. I.
Nov. 19, 1889—HERDMAN FRANK E..M. E. Hale Elevator Co., Calumet Bld'g, 189 La Salle St., Chicago, and La Grange, Ill.
May 27, 1885—HERMAN, LUDWIG....Cons. Eng. and Expert, 29 Euclid Ave. and 242 Bell Ave., Cleveland, Ohio.
May 21, 1884—HERRESHOFF, JOHN B.......Herreshoff Mfg. Co., Bristol, R. I.
April 7, 1880—HERRICK, J. A....Room 146 Kemble Bldg., 15 Whitehall St., New York City.
May 15, 1889—HERSHEY, MARTIN E.....Supt. F. & M. Dept., Harrisburg Car Mfg. Co., and 31 S. Third St., Harrisburg, Pa.
April 7, 1880—HEWITT, WM.†...Vice-Pres. and Eng'r Trenton Iron Co., Trenton, N. J.

* Manager, 1886-89. † Manager, 1884-87.

Nov. 1, 1883—HIBBARD, HENRY D....Supt. Hainsworth Steel Co, 25th and Smallman Sts., and 419 Penn. Ave., Pittsburgh, Pa., also West Roxbury, Mass.

Nov. 19, 1889—HIBBARD, THOMAS....M. E. and Ch'f D'ftsman Deane Steam Pump Works, and 232 Beach St., Holyoke, Mass.

Nov. 4, 1880—HIGGINS, MILTON P....Supt. Washburn Mach. Shop, and 228 West St., Worcester, Mass.

Nov. 2, 1882 } HIGGINS, SAMUEL.......Div. M. M., N. Y., P. & O. RR.,
May 27, 1885 } Meadville, Pa.

Nov. 29, 1887—HILDRUP, W. T....American Refrig. and Constrn. Co., Box 603, Harrisburg, Pa.

May 8, 1888—HILDRUP, W. T., JR......Supt. Car. Wheel Dept. Hbg. Car Mfg. Co., Box 603, Harrisburg, Pa

June 20, 1880—HILL, H. A.*.................156 Oliver Street, Boston, Mass.

May 21, 1884—HILL, JAMES W ...M. M & Car Builder, Peoria & Pekin Union R'y Co., Peoria, Ill.

May 21, 1884—HILL, WARREN E....V. P. Continental Iron Works, Brooklyn, N. Y.

Nov. 1, 1883 } HILL, WILLIAM..Asst. Supt. Collins Co., and P. O. Box 196,
May 15, 1889 } Collinsville, Conn.

May 8, 1888—HILLARD, CHARLES J....Mech. Eng., Box 282, Pittsburgh, Pa.

May 14, 1890—HILLES, T. ALLEN...Vice-Prest. The Hilles and Jones Co., Wilmington, Del.

June 13, 1883—HILLMANN, GUSTAV... Naval Architect, 470 Greene Ave., Brooklyn, N. Y.

Aug. 10, 1881—HOBBS, A. C....Supt. Union Metallic Cartridge Co., Bridgeport, Conn.

June 13, 1883—HOE, ROBERT....R. Hoe & Co., 504 Grand and Columbia Streets, New York City.

May 26, 1886—HOLLAND, JOHN....Agt. Cocheco Mfg. Co., and 113 Locust St., Dover, N. H.

April 7, 1880 } HOLLERITH, HERMAN ...Expert & Sol. of Patents,
June 13, 1883 } Room 48, Atlantic Bldg., Washington, D. C.

June 13, 1883—HOLLINGSWORTH, SUMNER....36 Federal Street, and 5 Fairfield Street, Boston, Mass.

May 21, 1884—HOLLIS, IRA N., Asst. Eng. U. S. N....Union I. W., San Francisco, Cal

April 7, 1880—HOLLOWAY, J. F.†..145 Broadway, Box 2227, and 230 West 59th St., New York City

Oct. 16, 1888—HOLLY, FRANK W....Eng. and Supt. Holly Mfg. Co., and 251 High St., Lockport, N. Y

Oct. 16, 1888—HOLMBOE, LEONHARD C. B....Chief D'tsman Illinois Steel Co., 1038 Rookery, and 4404 Cottage Grove Ave., Chicago, Ill.

April 7, 1880—HOLMES, ISAAC V.............................Beloit, Wis.

April 19, 1882—HORNIG, JULIUS L............7 Mandeville Avenue, Jersey City, N. J.

Nov. 30, 1886—HORNUNG, GEO..Civ. Hydr'c and Mech. Eng., 30 East 4th St., Cincinnati, Ohio, and 232 Fifth St., Newport, Ky.

May 31, 1887—HORTON, JAMES A....Supt. Eng. "Howard Metallic Brush Co.," 12 Pearl St., Boston, and Reading, Mass.

* Manager, 1885-88. † President, 1884-85; Manager, 1880-83.

Nov. 5, 1880—HOWARD, CHARLES P.... ....James L. Howard & Co., Hartford, Conn.
May 26, 1886—HOWE, HENRY M...............Cons. Metallurgist, 287 Marlborough St., Boston, Mass.
Nov. 19, 1889—HOWELL, EDWARD I. H....M. E., 220 S. 4th St., and 4636 Gmtn Ave., Phila., Pa.
April 19, 1882—HUGO, T. W........Gen. Mgr. Hartman Electric Co., and 221 6th Ave., W. Duluth, Minn.
May 15, 1889—HUGHES, E. W. M....Loco. and Car Dept., N. W. State R'y of India, and Fox Solid Pressed Steel Co., 1004 The Rookery, Chicago, Ill.
May 15, 1889—HUMPHREY, JOHN....Pres. and Gen. Mgr. Humphrey Mach. Co., and 400 Main St., Keene, N. H.
Nov. 6, 1884—HUMPHREYS, ALEX. C....Gen. Supt. United Gas Imp'm't Co., 813 Drexel Bldg., 5th and Chestnut Sts., and Chestnut Hill, Philadelphia, Pa.
Nov. 19, 1889—HUNT, ALFRED E......Pres. Pittsburgh Reduction Co., 95 5th Ave., and 272 Shady Ave., E. E., Pittsburgh, Pa.
Nov. 11, 1885—HUNT, CHARLES WALLACE (*Life Member*)....45 Broadway, New York City.
Nov. 30, 1886—HUNT, JOSEPH....Min. Eng. and Pres. Allentown F. & M. Co., 3d and Walnut Sts., Allentown, Pa.
April 7, 1880—HUNT, ROBERT W.*...Robert W. Hunt & Co., Insp. and Cons. Eng., 631 "The Rookery," Chicago, Ill.
May 14, 1890—HUNTER, GEO. E ...Assist. Supt. Elgin Nat'l Watch Co., and Watch St., Elgin, Ill.
May 15, 1889—HUNTER, J. S....Erection Dept. H. R. Worthington, 86 Liberty St., N. Y. City.
April 7, 1880—HUTTON, FREDERIC R..Adj. Prof. Mech. Eng. School of Mines Columbia Coll. and Secretary † of the Society, 12 W. 31st St., New York City.
May 27, 1885—HYDE, CHARLES E...Marine Eng'r Bath Iron Works, Bath, Me.

Nov. 6, 1884—IDE, ALBERT L....Prop'r and Eng. Ide Engine Works, Springfield, Ill.
May 15, 1889—IDELL, FRANK E.....Mech. Eng'r. 41 Dey St., New York City.
May 4, 1881—ILLINGWORTH, JOSEPH J....Chief Eng. Utica Steam Cotton Mills, Utica, N. Y.
May 27, 1885—INSLEE, WM. H..............50 Spruce Street, Newark, N. J.

May 27, 1885—JACOBI, ALBERT W.. Mech. Engr. Judson Pneumatic St. Ry. Co., 45 Broadway, New York City, and 286 S. 6th St., Newark, N. J.
May 15, 1889—JACOBUS, D. S....Asst. Prof. Experimental Shop W'k and Mechs., Stevens Institute, Hoboken, N. J.
May 14, 1890—JARVIS, SAMUEL E....Mech. Eng. and Mech. Supt. Jarvis Engine Co., 809 Vine St., Lansing, Mich.
May 27, 1885—JENKINS, JOHN...Milton Iron Works Rolling Mills, Milton, Pa.
Nov. 11, 1885—JENKINS, W. R....Jenkins & Lingle, Engrs. and Machsts., Bellefonte, Pa.

* Manager, 1882-85. † 1883 to 1890.

May 15, 1889—JENKS, WILLIAM H. . . . . Dynamic Eng'r, Brookville, Pa.
Nov. 19, 1889—JEWETT, L. C. . . . Erie City Iron Works, Erie, Pa., and 140 Warburton Ave., Yonkers, N. Y.
May 14, 1890—JOHNSEN, CARL A. . . . Chf. D'ftsman, H. R. Worthington Hydraulic Works, So. Brooklyn, N. Y.
May 21, 1884—JOHNSON, C. R. . . Pres. and Gen. Man. The Johnson RR. Signal Co., Rahway, N. J., and 155 West 58th St., N. Y. City.
April 7, 1880—JOHNSON, LEWIS. . . . Pres. Johnson I. W. Limited, P. O. Box 1200, and 85 Erato St., New Orleans, La.
May 14, 1890—JOHNSON, NILS. . . . Mech. Eng., 1458 Francis St., and 3,103 So. Jefferson Ave., St. Louis, Mo.
Nov. 19, 1889—JOHNSON, WARREN S. . . . . Eng. and Mgr. Johnson Elect. Service Co., 113 Clybourn St., and 2427 Cedar St., Milwaukee, Wis.
April 19, 1882—JOHNSON, WILLIAM. . . . . . . 637 F St., N. W. Wash'ngton, D. C.
Aug. 10, 1881—JONES, DAVID P. . . . Chf. Eng. U. S. N. Nav. Train'g Sta., Newport, R. I.
May 15, 1889—JONES, EDWARD H. . . . E. H. Jones & Co., Eng'rs and Contractors, 225 River St., and 69 Kennard St., Cleveland, O.
May 14, 1890—JONES, EDWIN H. . Prest. and Gen. Mgr. Vulcan Iron Wks., Wilkesbarre, Pa.
May 14, 1890—JONES, F. R. . . . Mech. and Elect. Eng., Western Eng. Co., Kearney, Nebraska.
May 4, 1881—JONES, HENRY C. . . . Hilles & Jones, 1315 Delaware Ave., Wilmington, Del.
May 14, 1890—JONES, JOHN T. . . . Mining Supt. Iron Mountain, Mich., and Prest. Bessemer Spike, Nail and Staple Co., 40 Dearborn St., Chicago, Ill.
April 7, 1880—JONES, WASHINGTON *. . . . Supt. Port Richmond I. W., 2257 Richmond St., and 1632 No. 15th Street, Philadelphia, Pa.
May 26, 1888—JONES, WM. H. . . Prest. and Gen. M'n'gr C. E. Jones & Bros. 28–32 Court St., and 62 Fulton Ave., W. H., Cincinnati, O.
Nov. 29, 1887—JONES, WILLIS C. . . . Mech. and Cons. Eng., Jones & Mack, 5 West 4th St., Cincinnati, O.
Nov. 29, 1887—JORDAN, SAMUEL S. . . . Hull D'tsman, The Wm. Cramp & Sons, S. & E. B. Co., 921 Spruce St., Phila., Pa.

Nov. 19, 1889—KAFER, JOHN C. . . . Passed Asst. Eng., U. S. N., Morgan Iron Works, 9th St., E. R., N. Y. City, and 38 E. 49th St., N. Y.
June 13, 1883—KAFFENBERGER, GUSTAV. . . . . . M. E. The Central Paper and Fiber Co., 171 Seneca St., and 183 Waverly Ave., Cleveland, O.
May 21, 1884—KANE, JOHN. . . . J. S. Graham & Co., and 260 Lyall Ave., Rochester, N. Y.
Nov. 30, 1886—KEBLER, JULIAN A . . . . Genl. Mgr. Colorado Fuel Co., 1657 Larimer St., Denver, Col.
May 15, 1889—KELLER, JOHN A. . M. E. and D'ftsman Black and Clawson Co., and 136 No. C St., Hamilton, Ohio,
May 27, 1885—KELLY, O. W. . . . . Supt. The O. S. Kelly Co., Springfield, O.

* Manager, 1880-81 ; Vice-President, 1881-1882.

May 26, 1886—KEMPSMITH, FRANK....Prest. Kempsmith Mach. Tool Co., and 881 Robinson Ave., Milwaukee, Wis.
May 31, 1887—KENT, EDMUND....M. M. Chapin Mine, Lock Box 23, Iron Mountain, Mich.
May 15, 1889—KENT, ELLIS C..Mgr. Bethlehem F. & M. Co., So. Bethlehem, Pa.
April 7, 1880—KENT, WM*........Cons. Eng., 125 *Times* Bldg., N. Y., and Passaic, N. J.
June 20, 1880 } May 15, 1889 } KEPPY, FREDERICK ...Supt. and Eng. Warner Bros., 79 Gregory St., Bridgeport, Conn.
May 26, 1886—KERR, WALTER C......Westinghouse, Church, Kerr & Co., 17 Cortlandt St., New York City, and 496 9th St., Brooklyn, N. Y.
June 13, 1883—KETTELL, CHARLES W... Geo. F. Blake Mfg. Co., Boston, and 3 Brewster Street, Cambridge, Mass.
May 26, 1886—KIMBALL, HIRAM....Mgr. F. B. Dept. Cleveland, City Forge and Iron Co., Cleveland, Ohio.
May 27, 1885—KINDER, J. J. DE...Cons. Eng., 901 Walnut St., and 626 N. 11th St., Phila., Pa.
Nov. 30, 1886—KING, CHAS. C....Supt. C. W. Hunt Co., P. O. Box 169, West New Brighton, Richmond Co., N. Y.
Aug. 10, 1881—KING, CHARLES I......Prof. Mech. Prac. Univ. of Wis., and Madison, Wis.
Nov. 5, 1880—KIRBY, FRANK E.............Eng. Kirby Bros., Detroit, Mich.
Nov. 2, 1882—KIRCHHOFF, CHARLES....*Iron Age*, 66 Duane St., N. Y., and 369 Union St., Brooklyn, N. Y.
Oct. 16, 1888—KIRK, WM. ADDISON LYLE..Ludlow Valve Co., 309 Fanning St., Chattanooga, Tenn.
May 26, 1886—KIRKEVAAG, PETER....Chief D'tsman and Mech. Eng., Wm. Tod & Co., Youngstown, Ohio.
Aug. 10, 1881—KLEIN, J. F....Prof. Mech. Eng. Lehigh University, P. O. Box 495, and 357 Market St., Bethlehem, Pa.
May 14, 1890—KNEASS, STRICKLAND L..Mech. Eng. with Sellers & Co., 1600 Hamilton St., and 2228 Pine St., Phila., Pa.
May 21, 1884—KNIGHT, CHAS. A....Babcock & Wilcox Co., 107 Hope Street, Glasgow, Scotland.
Nov. 19, 1889—KNOUS, FRANKLIN F......Mfr. Crochet Needles, Greystone, Conn.
Nov. 29, 1887—KRAUSE, ARTHUR..Arch't and Chf. D'tsman F. O. Matthiessen & Wiechers' Sugar Ref., Washington and Essex Sts., and 185 Summit Ave., Jersey City, N. J.

May 21, 1884—LADD, JAMES B......Mech. Eng. Penn Steel Co., Sparrows Point, Baltimore Co , Md.
Nov. 29, 1887—LAFORGE, FRED'K HENRY....Ch'f Insp. Conn. Mut. S. B. Insp. & Ins. Co., Waterbury, Conn.
May 15, 1889—LAIDLAW, WALTER.....V. P. and Mgr. Laidlaw & Dunn Co., Mfrs. Pumping Mach., Hydr. Mach., etc., Pearl and Plum Streets, Cincinnati, Ohio.
May 31, 1887—LAMBERT, WILBUR C..Mech. Eng'r, Room 7, Mitchell Bldg., 828 Chapel St., and 34 Clark St., New Haven, Conn.

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* Vice-President, 1888-90 ; Manager, 1885-88.

June 13, 1883—LANDRETH, OLIN H.*...Prof. Eng'g, Vanderbilt University, Nashville, Tenn.

May 26, 1886—LANE, HARRY M....Prest. Lane & Bodley Co., Cincinnati, Ohio.

April 19, 1882—LANE, J. S....Gen. Supt. M. C. Bullock Mfg. Co., Phœnix Bldg., Chicago, Ill.

Nov. 30, 1886—LANPHEAR, O. A......Gen. Man. Robb Closet Co., Main and Keck Sts., and Station C, Cincinnati, Ohio.

Nov. 2, 1882—LANZA, GAETANO....Prof. Theor. & App. Mech., Dept. Mech. Eng'g, Mass. Institute of Techn., Boston, Mass.

May 14, 1890—LAPE, W. E... Mech. Eng. Porter Mfg. Co., Ltd., and 407 Turtle St., Syracuse, N. Y.

May 8, 1888—LARKIN, FRED. A..Eng'r E. P. Allis & Co.,and 547 Jackson St., Milwaukee,Wis.

May 4, 1881—LAUREAU, LOUIS G..Metallurgical Eng., 60 Washington Sq. So., New York City.

Nov. 30, 1886—LAVAL, GEO. DE..Chf. D'ftsman Knowles Steam Pump W'ks, Warren, Mass.

Nov. 30, 1886—LAVERY, GEO L......M. E. Yale & Towne Mfg. Co., 152 & 154 Wabash Ave., Chicago, Ill.

May 27, 1885—LAWRANCE, J. P. STUART, P. A. E., U. S. N....Navy Yard, Norfolk, Va.

April 7, 1880—LEAVITT, E. D.†...2 Central Square, and 317 Harvard St., Cambridgeport, Mass.

May 31, 1887—LEAVITT, FRANK M......V. P. E. W. Bliss Co. W'ks, 17 Adams St., Brooklyn, N. Y.

April 19, 1882—LEMAN, WILLIAM T.....Supt. Acid, Wax and Candle Depts., Tide Water Oil Co., P. O. Box 220, and Ave. A and Linden St., Bayonne, N. J.

May 14, 1890—LEONARD, ARTHUR G.......Asst. to Supt. M. P. and Rolling Stock, N. Y. C. & H. R. RR. Co., Room 14, Grand Central Depot, N. Y. City.

April 7, 1880—LE VAN, W. BARNET....Consulting Eng'r, 3607 Baring Street, Philadelphia, Pa.

May 31, 1887—LEWIS, JAMES F....Rand Drill Co., 23 Park Place, and 128 West 59th St., New York City.

May 21, 1884—LEWIS, JOHN L....Ch'rman & Gen. Mgr. of Lewis Foundry & Machine Co., and Craig St., Pittsburgh, Pa.

May 14, 1890—LEWIS, ROLLIN C. W..Supt. Springer Torsion Bal. Co., 147 8th St. ,and 14 W. Hamilton Pl., Jersey City, N. J.

May 21, 1884—LEWIS, WILFRED..Asst. Eng. for Wm. Sellers & Co. (Icptd.), and 3234 Powelton Ave., Philadelphia, Pa.

May 26, 1886—LIEB, JNO. W., JR....Chf. Eng. Edison E. L. Co., 4 Via Sta. Radegonda, Milan, Italy.

June 13, 1883—LIPE, CHARLES E..........Mfr., 111 Holland Street, Syracuse, N. Y.

May 27, 1885—LIVERMORE, CHARLES W.....Middlesborough, Ky., and Athletic Club, 104 W. 55th St., New York City.

May 15, 1889—LOCKE, WARREN S.... Prof. Mechanics, R. I. Sch. of Design, 6 Pallas Street, Providence, R. I.

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* Vice-President, 1885-86. † President, 1882-83.

May 14, 1890—LODGE, WM.....Vice-Prest. & Gen. Mgr. Lodge & Davis Machine Wks., 6th St. & Eggleston Ave., Cinn., O.
June 20, 1880—LOGAN, WM. J...Logan Iron Works, Commercial & Clay Sts., Brooklyn, N. Y.
April 19, 1882—LORING, CHARLES H.*...Chief Eng. U. S. N., Navy Yard, N. Y., and 239 Clermont Ave., Brooklyn, N. Y.
Nov. 30, 1886—LUDLAM, JOS. S.......Agt. Merrimac Mfg. Co., Lowell, Mass.
April 7, 1880—LYNE, LEWIS F.... M. E. Jas. Beggs & Co., 9 Dey St., and 307 Grove St., Jersey City, N. J.

Nov. 30, 1886—MCBRIDE, JAMES....Supt. N. Y. Dye Wood Extract and Chemical Co., 146 Kent St., Brooklyn, N. Y.
May 15, 1889—MCCLATCHEY, A. F....Chief D'ftsman Wabash R'lw'y, Lock Box 676, Springfield, Ill.
May 15, 1889—MCDUFFIE, CHARLES D........Agent Manchester Mills, Manchester, N. H.
May 4, 1881—MCELROY, SAMUEL..Consulting Eng'r, Room 39, 170 Broadway, N. Y. City, and 50 Johnson Street, Brooklyn, N. Y.
Nov. 2, 1882—MCEWEN, J. H..............................Ridgway, Pa.
May 8, 1888—MCRAE, JOHN D.............Supt. Morris Machine Wks., Baldwinsville, N. Y.
Nov. 1, 1883—MACFARLAND, WALTER M., U. S. N....Bu. Steam Eng'g Navy Dept., Washington, D. C.
May 14, 1890—MACK, JOHN G. D....Mech. Eng. Jones & Mack, 5 W. 4th St., and 119 W. 5th St., Cinn., O.
May 14, 1890—MCKINNEY, ROBT. C.....Sec'y & Treas. Niles Tool Works, Hamilton, Ohio.
Nov. 1, 1883—MACKINNEY, WM. C..Supt. H. W. Butterworth & Sons, York and Cedar Sts., and 2316 E. York Street, Philadelphia, Pa.
May 21, 1884—MAGRUDER, WM. T....Prof. Mech. Eng., Vanderbilt Univ., Nashville, Tenn.
May 15, 1889—MAHON, WM. L.'E....D'ftsman and Mech. Eng. Frontier Iron and Brass Works, and 254 Jos. Campan Ave., Detroit, Mich.
May 21, 1884—MAHONY, JAMES.......Mech. Eng., 245 Broadway, New York City.
May 31, 1887—MAILLEFERT, GUSTAVE JACQUES..Supt. Erection N. Y., N. H. & H. R. R., and 245 Hallock Ave., New Haven, Conn.
Nov. 11, 1885—MAIN, CHARLES T..Supt. Lower Pacific Mills, Lawrence, Mass.
Oct. 16, 1888—MANCHESTER, ALFRED E....C., C., C. & I. R.R. Co., Fort Wayne, Ind.
May 21, 1884—MANNING, CHAS. H..P. A. Eng. U. S. N., Supt. Amoskeag Mills, Manchester, N. H.
May 31, 1887—MANSFIELD, ALBERT K..Asst. Supt. Buckeye Eng. Co., Salem, O.
May 14, 1890—MARGEDANT, WM. C..... Prest. & Mgr. Bentel & Margedant Co., Hamilton, O.
May 27, 1885—MARTENS, FERDINAND........India Rubber Comb Co., College Point, L. I.
May 14, 1890—MARTINEZ, MANUEL J..Mech. Eng., 115 Broadway, N. Y. City.
Nov. 5, 1880—MASON, WILLIAM....Supt. Eng'r Winchester Repeating Arms Co., and 15 Dixwell Ave., New Haven, Conn.

* Vice-President, 1885-87.

Nov. 1, 1883—MATLACK, DAVID J....Supt. Foundry, I. P. Morris & Co., and Langhorne, Bucks Co., Philadelphia, Pa.
Nov. 2, 1882—MATTES, WILLIAM F.....Gen. Mgr. West Superior Iron & Steel Co., West Superior, Wis.
May 15, 1889—MATTICE, ASA M....P. A. E. U. S. N., 2 Central Square, Cambridgeport, Mass.
Aug. 10, 1881—MAY, DE COURCY..I. P. Morris & Co., and 1230 Spruce Street, Philadelphia, Pa.
May 14, 1890—MELLEN, EDWIN D..Supt. Curtis, Davis & Co., 184 Broadway, and 31 Essex St., Cambridgeport, Mass.
May 14, 1890—MELLIN, CARL J....Chf. Dftsman. Richmond L. & M. Works, and 610 E. Leigh St., Richmond, Va.
April 7, 1880—MELVIN, DAVID N.........Linoleumville, Staten Island, N. Y.
May 4, 1881—MERRICK, J. VAUGHAN *....... Roxborough, Philadelphia, Pa.
Nov. 19, 1889—MESERVE, JOHN W....M. E. Yale & Towne Mfg. Co., Stamford, Conn.
Nov. 1, 1883—MESSIMER, HILLARY ...Supt. Mo. P. Calumet and Hecla Mining Co., Calumet, Mich.
April 7, 1880—METCALF, WM †..... Miller, Metcalf & Parkin, Pittsburgh, Pa.
May 26, 1886—METCALFE, HENRY........Capt. of Ordnance, U. S. A., West Point, N. Y.
May 31, 1887—METCALFE, JOHN....Mech. Eng., D'tsman, and Pat. Expt., Woonsocket, R. I.
May 31, 1887—MEYER, J. G. ARNOLD.......Mech. Eng. *Amer. Macht.*, 96 Fulton St., N. Y., and 122 Fair St., Paterson, N. J.
May 26, 1886—MIDDLETON, HARVEY...Supt. Mo. P. and Mach. U. P. R.R., Omaha, Neb.
Nov. 29, 1887—MILES, FREDERICK B....Bement, Miles & Co., 21st and Callowhill Sts., and 1718 Walnut St., Philadelphia, Pa.
May 4, 1881—MILLER, ALEXANDER....Brown & Miller, Hudson and Morris Sts., Jersey City, N. J.
May 14, 1890—MILLER, FRED. J... ...Associate Editor *American Machinist*, 96 Fulton St., N. Y. City.
May 15, 1889—MILLER, JAMES S....Mech. Eng. & Chief D'fsman Stearns Mfg. Co., and 927 E. 21st St., Erie, Pa.
June 13, 1883—MILLER, LEBBEUS B....Supt. The Singer Mfg. Co., and 1025 E. Jersey St., Elizabeth, N. J.
June 13, 1883—MILLER, LEWIS...........Aultman, Miller & Co., Center St. R'y Crossing & Oak Pl., Akron Ohio.
May 8, 1888—MILLER, T. SPENCER........Mech. Eng., 96 Liberty St., New York City, and 235 Quincy St., Brooklyn, N. Y.
May 26, 1886—MILLER, WALTER........Mech. Eng. Globe Iron Works, 85 Liberty St., Cleveland, Ohio.
May 26, 1886—MINOT, H. P....Prop. Industrial Mach. W'ks, and 56 Vine St., Columbus, Ohio.
May 21, 1884—MIRKIL, THOMAS H., JR...Mech. Eng., Southwark F. & M. Co., 430 Washington Ave., and 838 No. 40th St., Philadelphia, Pa.
Nov. 1, 1883—MOFFAT, EDWARD S..Gen. Man. Lackawanna Iron and Coal Co., and 306 Quincy Ave., Scranton, Pa.

* Vice-President, 1883-85. † Vice-President, 1882-84.

May 26, 1886—MOHR, LOUIS....Cons. Eng'r, John Mohr & Son,
32 to 42 Illinois Street, Chicago, Ill.
May 26, 1886—MONAGHAN, WM. F....Eng. Dept. U. S. Ill. Co.,
Times Bldg., and 169 E. 69th St., N. Y. City.
May 15, 1889—MONTGOMERY, H. M.....Chief D'ftsman Mach. Dept.
N. Y. and N. E. R. R., Norwood, Mass.
May 14, 1890—MOORE, A. G...Supt. & Engineer Cinn. Water Wks., Cinn., O.
May 14, 1890—MOORE, D. G........Vice-Prest. S. L. Moore & Sons Co.,
"Crescent Iron Wks.," Front, Marshall & Franklin Sts.,
Elizabeth, N. J.
May 26, 1886—MOORE, ENOS L.........Eng'r and Contractor, Portsmouth, O.
Nov. 2, 1882—MOORE, JOHN W....Chief Engineer U. S. N., Navy Yard,
Mare Island, Cal.
May 27, 1885} MOORE, W. J. P..Worthington Pumping Engine Co.,
Oct. 16, 1888} 153 Queen Victoria St., London, E. C., England.
May 26, 1886—MORAVA, WENSEL....Supt. Bouton Foundry Co.,
39th and Winter St., and 5621 Monroe Ave., Chicago, Ill.
May 8, 1888—MOREAU, EUGENE....Cons. Eng. Siemens-Lungren Co., 21st St.
and Washington Ave., and Hotel Lafayette, Philadelphia, Pa.
May 4, 1881—MORGAN, CHARLES H.*...Eng. and Contr., 25 Lincoln St.,
and 28 Catharine St., Worcester, Mass.
Aug. 10, 1881—MORGAN, JOSEPH, JR.†...Chf. Eng. Cambria Iron Works,
and 10 Morris St., Johnstown, Pa.
May 21, 1881—MORGAN, JAMES.... ..Supt. Structural Dept. American Iron &
Steel Co., and 2204 Carson St., Pittsburgh, Pa.
Nov. 2, 1882—MORGAN, THOMAS R., SR.‡.........President and Treas.
Morgan Eng'g Co., Alliance, Ohio.
May 21, 1884—MORGAN, T. R., JR....Sec'y. & Gen. Mgr. Morgan Eng'g Co.,
Alliance, Ohio.
Nov. 2, 1882—MORRIS, HENRY G.§...Eng'r and Mach't, 926 Drexel Bldg.,
Philadelphia, Pa.
May 26, 1886—MORRISON, WM. A....Chief Eng'r, Lowell Mfg. Co.,
P.O. Box 373, Lowell, and 9 Felton St., Cambridge, Mass.
May 21, 1884—MORSE, CHAS. JAS....Morse Bridge Co., Youngstown, O.,
and 821 27th St., Denver, Col.
May 21, 1884—MORSE, CHAS. M.....Mech. Eng., 228 Pearl St., and
292 Franklin St., Buffalo, N. Y.
April 7, 1880—MORTON, HENRY‖...Pres. Stevens Inst. Tech., Hoboken, N. J.
May 27, 1885—MUCKLÉ, M. RICHARDS, JR....Mech. Eng., 212 Drexel Bldg.,
and 1722 Pine St., Philadelphia, Pa.
May 26, 1886—MULLEN, JOHN....Propr. and Gen. Man. Shamokin I. W.,
Shamokin, Pa.
May 14, 1890—MULLER, EDWARD A...Vice-Prest. and Supt. The Bradford
Mill Co., Cinn., O.
Nov. 11, 1885—MÜLLER, MAURICE A....Chf. D'tsman and Mech. Eng.
U. S. E. I. Co., Weston Factory, and 451 Broad St.,
Newark, N. J.
May 8, 1888—MÜLLER, TEILE H..Cons. Eng., Geo. M. Newhall Eng'g Co.,
136 So. 4th St., and 1238 So. Broad St., Phila., Pa.

* Manager, 1884-87. † Vice-President, 1886-88. ‡ Manager, 1886-88.
§ Vice-President, 1887-89. ‖ Vice-President, 1882-84.

Nov. 29, 1887—MUNCASTER, WALTER JAMES....Supt. F'dry and Mach. Works, and Queen City Hotel, Cumberland, Md.
Nov. 4, 1880—MURPHY, EDWARD J....Mech. & Consult'g Eng. Hartford S. B. Insp. & Ins. Co., and 15 Capitol Ave., Hartford, Conn.
June 13, 1883—MURRAY, S. W........Murray, Dougal & Co., Milton, Northumberland Co., Pa.

Nov. 4, 1880—NAGLE, A. F....Mech. Eng., Prest. Nagle Auto. Sprinkler Co., 115 Dearborn Street, Chicago, Ill.
April 7, 1880—NASON, CARLETON W.*....Prest. Nason Mfg. Co., 71 Beekman St., and 136 E. 16th St., N. Y. City.
Nov. 19, 1889—NAYLOR, ERNEST W....Cons. Eng., 149 Broadway, N. Y. City.
Nov. 29, 1887—NAYLOR, JOHN SAMUEL ...Walker Mfg. Co., Waverly Ave. and Breakwater St., Cleveland, O.
May 14, 1890—NEAVE, JOS. S....Sec'y. and Treas. The Walton Architectural Iron Co., Cinn., O.
Nov. 1, 1883—NEWCOMB, CHAS. L....Supt. Deane Steam Pump Co., and 252 Pine St., Holyoke, Mass.
May 14, 1890—NICHOLS, FRANK L....Mech. Eng. and Dftsman. Pneum. Dyn. Gun Co., Room 69, 71 Broadway, N. Y. City.
Oct. 16, 1889—NICHOLSON, DAVID KIRK....Asst. Supt. Roll'g Mills, Penn. Steel Co., Lock Box 9, Steelton, Pa.
Nov. 2, 1882—NICHOLSON, STEPHEN..V. P. and Treas. Nicholson & Waterman, Mason and Beverly Sts., and 12 George St., Providence, R. I.
April 7, 1880—NICHOLSON, W. T...Prest. Nicholson File Co., and 17 Brownell St., Providence, R. I.
Nov. 19, 1889—NICOLL, CHAS. H....Chf. Eng. Ballantine's Brewery, and 289 High St., Newark, N. J.
May 31, 1887—NICOLLS, JASPER OLIVER.... W. J. Nicolls & Co., 216 So. Third St., Phila., Pa.
Nov. 11, 1885—NICOLLS, WM. J....Treas. Irvona Coal Co., 216 S. 3d St., Philadelphia, Pa.
June 20, 1880—NORMAN, GEO. H. (*Life Member.*)....343 Beacon St., Boston, Mass.
June 13, 1883—NORTON, HAROLD P....Asst. Eng. U. S. N., Bureau of Steam Engineering, Navy Dept., Washington, D. C.

Nov. 29, 1886—O'CONNELL, JOHN C....Prop'r Cotton Compress, and 110 Moulton St., Montgomery, Ala.
April 7, 1880—ODELL, WM. H....Mech. Eng'r, 125 Buena Vista Ave., Yonkers, N. Y.
May 14, 1890—OTIS, SPENCER....Gen. Mgr. Kansas City Switch & Frog Co., 305 Mercantile Bldg., Kansas City., Mo.
May 31, 1887—OWEN, FREDERICK N ...Civ. and San. Eng., 13 William St. N. Y. City.

Nov. 30, 1886—PACKARD, L.... Master Car Builder N. Y. C. & H. R. RR. Car Shops, and N. Y. C. Ave., West Albany, N. Y.
June 13, 1883—PALMER, GEORGE E..........64 So. Canal Street, Chicago, Ill.
June 13, 1883—PANKHURST, JOHN F....Gen. Man. Globe Iron W'ks, Centre and Spruce Sts., and 202 Franklin Ave., Cleveland, O.

* Manager, 1889-92.

Nov. 6, 1884—PANTALAONI, GUIDO....Westinghouse Elect. Co.,
Amer. Cent. Bldg., St. Louis, Mo.

May 15, 1889—PARK, WILLIAM R....Supt. and Mech. Eng. Park Mfg. Co.,
34 Beach St., Boston, Mass.

May 26, 1886—PARKER, CHAS. D...Crompton Loom W'ks and 110 Green St.,
Worcester, Mass.

Nov. 11, 1885—PARKER, CHARLES H.........Edward Kendall & Sons, and
302 Harvard St., Cambridgeport, Mass.

April 19, 1881—PARKER, WALTER EDWARD....Agt. Pacific Mills, and 217
Haverhill St., Lawrence, Mass.

April 7, 1880—PARKHURST, E. G.....Asst. Supt. Pratt & Whitney Co.,
and 50 Sumner St., Hartford, Conn.

Aug. 10, 1881—PARKS, EDWARD H....Mech. Eng. Brown & Sharpe Mfg. Co.,
Providence, R. I.

May 15, 1889—PARSONS, FREDERICK W....Supt. B. W. Payne & Sons,
and 351 Columbia St., Elmira, N. Y.

May 27, 1885 } PARSONS, HARRY DE B....Consult'g Eng.,
May 31, 1887 } 35 Broadway, Room 70, N. Y City.

May 21, 1884—PARSONS, WILLARD P...Walter A. Wood M. & R. Machine Co.
Hoosick Falls, N. Y.

Aug. 10, 1881—PARTRIDGE, WM. E..Insp. 2d Dist. B'd of Health, 42 Bleecker
St., New York City, and Passaic Bridge, N. J.

Oct. 16, 1888—PASSEL, GEORGE W...Chf. D'tsman and Des., J. A. Fay & Co.,
Cincinnati, O.

Nov. 2, 1882—PATTON, WILLIAM HENRY....Supt. Broken Hill Mines,
Broken Hill, N. S. Wales, Australia.

April 19, 1882—PAYNE, DAVID W..............Payne Eng. Co., Elmira, N. Y.

May 21, 1884—PEABODY, CECIL H.....Associate Prof. Steam Eng'g Mass.
Inst. Tech., Boston, Mass.

Nov. 19, 1889—PEARSON, F. S....Chf. Eng. Elect. Dept. West End St. R'y,
81 Milk St, Boston, Mass.

Nov. 29, 1887—PEARSON, WM. ANSON, JR..Supt. Boies Steel Car Wheel Wks.,
620 Adams Ave., Scranton, Pa.

Nov. 30, 1886—PEARSON, W. B....Man'g'r in Chicago A. L. Ide & Son,
25 Third Ave., Chicago, Ill.

May 14, 1890—PECK, STAUNTON B....Assist. Chf. Eng. Link Belt Eng'g Co.,
Nicetown, Phila., Pa.

Nov. 29, 1887—PEIRCE, WM. HENRY.....Supt. Marine Installations, United
Edison Mfg. Co., 65 Fifth Ave., New York City.

Nov. 19, 1889—PENDLETON, J. H....Pendleton & Thomas, Expts. and Cons.
Engs., 121 Nassau St., N. Y. City, and 601 Hamburg
Ave., Brooklyn, N. Y.

May 21, 1884—PENDRY, WM. ALLEN..Hoffman Mach. Co., 65 Farnsworth
Street, Detroit, Mich.

April 7, 1880—PENNEY, EDGAR....Gen. Supt. Frick & Co., Waynesboro', Pa.

May 15, 1889—PENRUDDOCK, J. H....Chf. D'fsman Loco. Shops
C. & G. T. R'l'y, Fort Gratiot, Mich.

May 27, 1885—PERKINS, GEORGE H...Atlantic Ref. Co., 413 So. Broad Street,
Philadelphia, Pa.

May 14, 1890—PERRY, NELSON W........Elect. Eng., 238 Auburn Ave.,
Cinn., O.

April 7, 1880—PERRY, WM. A....H. R. Worthington, 86–88 Liberty Street, and Box 2227, New York City.

May 27, 1885—PHILIPS, FERDINAND....Philips Townsend Co., No. Penn Junction, and 505 North 21st St., Philadelphia, Pa.

Aug. 10, 1881—PHILLIPS, FRANKLIN....Hewes & Phillips Iron Works, Newark, N. J.

Aug. 10, 1881—PHILLIPS, GEORGE H......Mech. Eng. Hewes & Phillips Iron Works, Newark, N. J.

April 7, 1880—PICKERING, THOMAS R......Pickering Governor Co., Portland, Conn.

Oct. 16, 1888—PIERCE, NORMAN M..........Prest. West Nashville Ry., 39 Baxter Court, Nashville, Tenn.

Nov. 2, 1882—PITKIN, A. J....Schenectady Loco. Works, Schenectady, N. Y.

June 13, 1883—PITKIN, JULIAN H...M. M. Wm. Deering & Co., Chicago, Ill., & P. O. Box 158, Ravenswood, Ill.

Nov. 30, 1886—PLAMONDON, AMBROSE....A. Plamondon Mfg. Co., 57–67 So. Clinton St., Chicago, Ill.

Nov. 30, 1886—PLAMONDON, CHAS. A.......A. Plamondon Mfg. Co., 57–67 So. Clinton St., Chicago, Ill.

Nov. 19, 1889—PLATT, GEO. H....Foreman Mach'y Harlem River Shops, N. Y., N. H. & H. R. R., and 677 E. 134th St., N. Y. City.

May 15, 1889 }
May 14, 1890 } PLATT, JOHN.Cons. Eng., High Orchard Road, Gloucester, Eng.

April 7, 1880—POND, FRANK H.....Prest. Pond Eng'g Co., 709 Market St., St. Louis, Mo.

May 14, 1890—PONTEN, ANDERS....Mech. Eng., 35 William St., Room 26, and 17 W. 30th St., N. Y. City.

May 15, 1889—POORE, TOWNSEND....M. E. Coal Dept. D. L. & W. RR. Co., and 1730 Capouse Ave., Scranton, Pa.

Nov. 30, 1886—POPE, SAMUEL I..S. I. Pope & Co., Steam Heating, 28 and 30 Market St., and 793 W. Monroe St., Chicago, Ill.

Nov. 4, 1880 }
May 21, 1884 } PORTER, HOLBROOK F. J....Supt. Bldgs. & Grds., Columbia College, 41 East 49th Street, New York City.

Oct. 16, 1888—PORTER, O. S..Propr. O. S. Porter's Cotton Mills, Covington, Ga.

May 14, 1890—PORTERFIELD, HENRY A....Engineer of Tests, Carnegie Bros. & Co., Braddock, Pa., and 315 So. Hiland Ave., Pittsburgh, E. E., Pa.

Aug. 10, 1881—POSSONS, N. S....Supt. Brush Electric Co., and 615 Case Ave., Cleveland, Ohio.

May 26, 1886—POST, JOHN, JR....Mech. Eng'r, 70 Kilby Street, Boston, Mass.

May 27, 1885—POTTER, CHARLES......Power Printing Presses, and 43 W. 7th St., Plainfield, N. J.

May 15, 1889—POTTER, FREDERICK D......Williams & Potter, 15 Cortlandt St., and 173 W. 83d St., N. Y. City.

April 7, 1880—POWEL, SAMUEL W..........433 No. 6th St., Hamilton, Ohio.

April 7, 1880—PRATT, FRANCIS A.*..The Pratt & Whitney Co., Hartford, Conn.

April 7, 1880—PRATT, NAT. W........Treas. The Babcock & Wilcox Co., 30 Cortlandt Street, New York City.

May 26, 1886—PRENTICE, LEON H....Pres't L. H. Prentice Co., 203 and 205 Van Buren St., Chicago, and Waukegan, Ills.

* Manager, 1880–81; Vice-President, 1881.

May 31, 1887—PRENTISS, FRED. H....Manager and Eng'r N. Y. Steam Co., 2 Cortlandt St., N. Y. City.
May 14, 1890—PRICE, J. A.......Prest. Scranton Stove Works, Scranton, Pa.
May 26, 1886—PRINDLE, EDWARD T.............V. P. and Supt. Prindle Mfg. Co., Aurora, Ill.
Nov. 5, 1880—PUSEY, CHAS. W........Gen. Man. The Pusey & Jones Co., and 1112 Washington St., Wilmington, Del.

May 31, 1887—RADFORD, BENJ. F......Gen. Man. Am. Tool & Mach. Co., 84 Kingston St., Boston, Mass.
May 21, 1884—RAE, CHAS. WHITESIDE....P. A. Eng. U. S. N., Office Naval Intelligence Bureau of Nav., Navy Dept, Washington, D. C.
Nov. 30, 1886—RAMSAY, H. ASHTON........Consult'g Eng'r, 406 Second St., Baltimore, Md.
May 14, 1890—RAMSAY, JAS. D....Asst. Supt. Mo. P. Calumet and Hecla Mine, Calumet, Mich.
May 21, 1884—RANDOLPH, LINGAN STROTHER.......Eng'r of Tests, B. & O. R.R., Mt. Clare, and 216 W. Madison St., Baltimore, Md.
Nov. 6, 1884—RAYNAL, ALFRED H..........Gen. Supt. Richmond L. & M. Works, Richmond, Va.
April 7, 1880—REED, EDWARD M............Vice-President, N. Y., N. H. & H. R. R. Co., New Haven, Conn.
May 14, 1890—REISS, GEORGE T....Chief D'ftsman Niles Tool Works, Hamilton, O.
Aug. 10, 1881—REMICK, OLIVER P......First Asst. Eng. U. S. Rev. Marine, U. S. S. "Gallatin," Boston, Mass.
Nov. 2, 1882—RENWICK, EDWARD S...Mech. Eng. and Exp't, 19 Park Place, New York City.
May 21, 1884—REYNOLDS, EDWIN........Gen. Supt. Reliance Works, E. P. Allis & Co., Milwaukee, Wis.
May 14, 1890—RICE, ALVA C..Gen. Supt. and Cons. Eng. Stillwell & Bierce Mfg. Co., and 324 W. 4th St., Dayton, Ohio.
May 14, 1890—RICE, RICHARD H....Chf. D'ftsman, E. D. Leavitt, 2 Central Sq., and 5 Ellsworth Ave., Cambridgeport, Mass.
April 7, 1880—RICHARDS, CHARLES B.*....Prof. Mechan. Eng'g Sheff. S. S., Yale Univ., and 313 York St., New Haven, Conn.
April 7, 1880—RICHARDS, FRANCIS H...Eng'g and Pat. Offices, 253 Main St., Hartford, Conn.
May 15, 1889—RICHARDS, GEO............Broadheath, Manchester, England.
May 15, 1889—RICHMOND, GEORGE.........Mech. Eng., 2296 Seventh Ave., New York City.
May 14, 1890—RICHTER, ERNST....Chf. D'ftsman The G. A. Gray Co., and 487 Sycamore St., Cinn., O.
May 31, 1887—RIDGWAY, J. T..........V.-P. Star Rubber Co., Trenton, N. J.
May 14, 1890—RIESENBERGER, ADAM........Assist. Prof. Mech. Drawing, Stevens Inst. Tech., Hoboken, and 44 Union Pl., Town of Union, N. J.
Oct. 16, 1888—ROBB, DAVID WENTWORTH....Mech. Supt. A. Robb & Sons, F. & M. Works, Amherst, N. S.

* Vice-President, 1888-90; Manager, 1880-82.

Nov. 19, 1889—ROBERTS, E. P......Supt. and Elect. Swan Mfg. Co., Belden St., and 32 Olive St., Cleveland, Ohio.
May 8, 1888—ROBERTS, GEORGE J....Eng. The United Gas Imprt. Co., 812 Drexel Bldg., Phila., Pa.
Nov. 30, 1886—ROBERTS, FRANK C....Civ. and Mech. Eng., Brown Building, Fourth and Chestnut Sts., Room 20, and 3223 Powelton Ave., Philadelphia; also Lewis Block, Pittsburgh, Pa.
May 8, 1888—ROBERTS, PERCIVAL JR., Supt. Pencoyd Iron W'ks, Pencoyd, Pa.
May 8, 1888—ROBERTS, WILLIAM.............Paper Mfgr., Waltham, Mass.
May 31, 1887—ROBERTS, THOMAS HERBERT.......Mech. Supt. C. & G. T. and D. G. H. & M. R'ys, and 63 Alexandrine Ave., Detroit, Mich.
May 8, 1888—ROBERTSON, R. A., JR.......Treas. Bldr's. Iron Fdry., 22 Codding St., Box 218, Providence, R. I.
June 13, 1883—ROBINSON, A. WELLS......Mech. Eng., Bucyrus St'm Shovel and Dredge Co., Bucyrus, O.
April 7, 1880—ROBINSON, G. H.......15 Cortlandt St., and 339 W. 57th St., New York City.
May 27, 1885—ROBINSON, J. M....M. M. and Supt. Mills Bldg., 15 Broad St., New York City.
April 7, 1880—ROBINSON, S. W.*.......Prof. Mech. Eng. State University, and 1353 Highland St., Columbus, Ohio.
June 13, 1883—ROBY, LUTHER A.............Otis Steel Co., Cleveland, Ohio.
Nov. 6, 1884 } ROCHE, JOHN A... Man. J. A. Fay & Co., 207-9 Lake St., and
Nov. 30, 1886 } 433 Warren Ave., Chicago, Ill.
Nov. 19, 1889—ROELKER, H. B...........Mech. Eng., 22 Cortlandt St., and 543 E. 86th St., N. Y. City.
May 21, 1884—ROGERS, WM. A...Prof. Physics, Colby Univ., Waterville, Me.
May 8, 1888—ROGERS, WINFIELD S........Supt. Estate F. W. Richardson, 464 Eighth St., Troy, N. Y., and 1408 Third Ave., West Troy, N. Y.
May 27, 1885 } ROMMEL, CHARLES E......Asst. Eng. U. S. N., Navy Dept.,
Nov. 11, 1885 } Washington, D. C.
May 15, 1889—RONEY, W. R..Mech. Eng. Westinghouse, Church, Kerr & Co., 158 Lake St., Chicago, Ill.
May 26, 1886—ROOD, VERNON H..........Supt. Jeanesville I. W., Jeanesville, Luzerne Co., Pa.
Nov. 4, 1880—ROOT, WM. J....Mech. Eng'r, H. R. Worthington, 86 Liberty St., New York City.
April 7, 1880—ROSE, JOSHUA.............Girton Lodge, Park Road, East Twickenham, England.
April 19, 1882—ROWLAND, THOS. F. (*Life Member*)..Prest. Continental I. W., Station "G," Brooklyn, N. Y., and 329 Madison Ave., N. Y. City.
May 21, 1884—ROWLAND, THOS. F., JR......Sec. & Treas. Continental Iron Works, and Station "G," and 170 Keap St., Brooklyn, N. Y.
May 15, 1889—ROYCE, HARRISON A.....Gen. Mgr. Thomson Elect. Welding Co., 89 State St., Room 40, Boston, Mass.
May 27, 1885—RUMELY, W. N..V.-P. and Supt. M. Rumely Co., La Porte, Ind.
Nov. 29, 1887—RUSSEL, WALTER S........V.-P. and Mgr. Russel Wheel and Fdy. Co., foot of McDougall Ave., and 863 Jefferson Ave., Detroit, Mich.

* Manager, 1881-84.

Nov. 30, 1886—RUTH, WILBUR M.....2 Belvidere, cor. 8th and Grand Aves., Milwaukee, Wis.
May 15, 1889—RUUD, EDWIN......Fuel, Gas and Elect. Eng. Co., Westinghouse Bldg., Pittsburgh, Pa.

Nov. 29, 1887—SAGUE, JAMES EDWARD.......Div. M. M. N. Y., L. E. & W. RR., Rochester, N. Y.
May 21, 1884—SANCTON, EDWARD K.......Supt. Penn Ave. Works Dickson Mfg. Co., Scranton, Pa.
May 8, 1888—SANDERS, NEWELL..........Prest. & Mgr. Chatta. Plow Co., Chattanooga, Tenn.
Nov. 30, 1886—SANGUINETTI, PERCY A....2128 No. 11th St., Philadelphia, Pa.
May 26, 1886—SARGENT, JOHN W...........Chief D'ftsman, Dickson Mfg. Co., Scranton, Pa.
June 13, 1883—SCHEFFLER, F. A......Gen. Supt. Westinghouse Elect. Co., Pittsburgh, Pa.
May 27, 1885—SCHLEICHER, ADOLPH W..M. E. Schleicher, Schumm & Co., 3313 Walnut Street, Philadelphia, Pa.
May 21, 1884—SCHUHMANN, GEORGE......Supt. Penn. Iron Works, 50th St. and Merion Ave., and 2422 Poplar St., Philadelphia, Pa.
Nov. 29, 1887—SCHUMANN, F.....Prest. and Gen. Mgr., Tacony Iron & Metal Co., Tacony, Phila., Pa.
May 31, 1887—SCHUTTE, LOUIS......Mech. Eng., 12th and Thompson Sts., and 1509 N. 17th St., Philadelphia, Pa.
May 21, 1884—SCHWAMB, PETER.........Assoc. Prof. Mechanism and Dir. Workshops Mass. Inst. Tech., Boston, also Arlington, Mass.
Oct. 16, 1888—SCHWANHAUSSER, WILLIAM.......Gen. Fmn. Hyd. Works, H. R. Worthington, So. Brooklyn, and 461 Eighth St., Brooklyn, N. Y.
Nov. 2, 1882—SCOTT, IRVING M....... Union Iron Works, P. O. Box 2128, San Francisco, Cal.
June 13, 1883—SCOTT, OLIN....Mech. Engineer and Prest. The Scott and Roberts Co., Wood Pulp Mach., and Prop. Benn. Mach. Works, Bennington, Vt.
May 31, 1887—SCOTT, WALTER W..D'ftsman Pneum. Dyn. Gun Co., 71 B'way, and 33 Ege Ave., Jersey City, N. J.
May 8, 1888—SCOVEL, MINOR......Pres. The Scovel and Irwin Const. Co., 9 Cole Bldg., Nashville, Tenn.
May 8, 1888—SCOVILLE, H. H..Prop'r Scoville I. W., 250-254 So. Clinton St., and 152 So. Sangamore St., Chicago, Ill.
May 15, 1889—SCRIBNER, CHARLES W...Prof. Mech. Eng. Coll. Agr. & Mech. Arts, Ames, Ia.
May 15, 1889—SEDERHOLM, EDWARD THEODOR..Mech. Eng. Fraser & Chalmers, and 107 So. Hoyne Ave., Chicago, Ill.
April 7, 1880—SEE, HORACE*....Eng'r and Naval Arch't, 1 Broadway and 48 W. 20th St., New York City.
April 7, 1880—SEE, JAMES W...................Mech. Eng'r, Hamilton, Ohio.
April 7, 1880—SELLERS, COLEMAN †........Prof. of Eng'g and Cons. Eng., 3301 Baring Street, Philadelphia, Pa.

* President, 1887-88. † President, 1885-86.

Nov. 2, 1882—SELLERS, COLEMAN, JR.. Asst. Man. Wm. Sellers & Co. (Incp.), 1600 Hamilton St., and 410 N. 33d Street, Philadelphia, Pa.
May 26, 1886—SELLERS, MORRIS *..............216 Phenix Bldg., Chicago, Ill.
April 7, 1880—SELLERS, WILLIAM.... 1600 Hamilton and 1819 Vine Street, Philadelphia, Pa.
Nov. 19, 1889—SEWALL, M. W.......Pneum. Dyn. Gun Co., 71 Broadway, Room 69, and 210 Waverley Place, New York City.
May 8, 1888—SHACKFORD, JAMES M..........Chief D'tsman, Mach. Dept. C. & A. R. R., Bloomington, Ill.
Nov. 6, 1884—SHARP, JOEL †..............Pres't. Buckeye Engine Co., and 84 Broadway, Salem, Ohio.
May 14, 1890—SHARPLES, P. M...... ....Westchester, Pa.
May 8, 1888—SHAW, T. JACKSON.... Supt. Eng. Harlan & Hollingsworth, and 1403 Van Buren St., Wilmington, Del.
May 14, 1890—SHELDON, F. P....Mill & Mech. Eng. & Arch., 65 Westminster St., and 47 Parade St., Prov., R. I.
May 21, 1884—SHELDON, THOS. C.......Supt. Lancaster Mills, Boylston, Mass.
Nov. 19, 1889—SHELMIRE, W. H., JR.......Pencoyd Iron Works, Pencoyd, Pa.
May 8, 1888—SHEPPARD, FRANK L....Supt. Mo. P. Penna. Div. P. R. R., Altoona, Pa.
May 14, 1890—SHIRRELL, DAVID....Dft'sman, Loco. Dept. Richmond L. & M. Wks., and 417 West Clay St., Richmond, Va.
April 7, 1880—SILVER, WM. J....Mech. Eng'r, 143 West No. Temple St., Salt Lake City, Utah.
Nov. 29, 1887—SIMONDS, GEO. F.........Prest. Simonds Roll'g Mach. Co., Fitchburg, Mass.
May 31, 1887—SIMPKIN, WILLIAM.......Simpkin & Hillyer, 1105 Main St., Richmond, Va.
May 26, 1886—SIMS, GARDINER C. (*Life Member*)....Gen. Mgr. Armington & Sims Eng. Co., Prov., R. I.
June 13, 1883—SINCLAIR, ANGUS....*National Car and Locomotive Builder*, Morse Bldg., 140 Nassau St., New York City.
Nov. 29, 1887 } May 14, 1890 } SINCLAIR, GEO. M.....Mech. Eng. Ordnance & Marine Forging Dept. Bethlehem Iron Wks., So. Bethlehem, Pa., and 3910 Chestnut St., Phil., Pa.
May 21, 1884—SKINNER, L. G.....................Skinner & Wood, Erie, Pa.
May 21, 1884—SMALL, H. J........Supt. Mo. P., and Mach. So. Pac. Ry., Sacramento, Cal.
Nov. 29, 1887—SMITH, A. P....Pat. Expt. & Sol., 606 Eleventh St., N. W., Washington, D. C.
May 27, 1885—SMITH, C. A........D'tsman and Des. Campbell Mach. Co., 171 Broad St., Pawtucket, R. I.
Nov. 19, 1889—SMITH, CHAS. F...Supt. U. S. Butter Extr. Co., cor. Orange and High Sts., and 116 Rector St., Newark, N. J.
May 31, 1887—SMITH, CHARLES H. L........Supt. Union Porcelain Works, 300 Eckford St., and 119 Milton St., Brooklyn, E. D., N. Y.
Aug. 10, 1881—SMITH, GEORGE H......Beaman & Smith, 117 Harrison St., and 28 Arch St., Providence, R. I.
May 4, 1881—SMITH, HORACE S............ Mgr. Illinois Steel Co., Joliet, Ill.

* Manager, 1887-90. † Vice-President, 1889-91.

June 13, 1883—SMITH, JESSE M..Mech. and El. Eng. and Exp't in Pat. Cases, 36 Moffat Block, and 51 Davenport St., Detroit, Mich.

May 4, 1881—SMITH, OBERLIN *........Pres't Ferracute Machine Company, and Lochwold, Bridgeton, N. J.

May 8, 1888—SMITH, SCOTT A..... Propr. Prov. Oil Co., and V.-P. Prov. Mut. S. B. Ins. Co., Box 1386, and 282 Greenwich St., Providence, R. I.

April 7, 1880—SMITH, SIDNEY L.......... Supt. Roxbury Carpet Co., and 37 Simmons Street, Boston, Mass.

May 26, 1886—SMITH, T. CARPENTER....Mech. Eng'r, 212 Drexel Bldg., 3303 Hamilton St., Philadelphia, Pa.

May 14, 1890—SMITH, THOS. G., Jr ...Mech. Eng., care of Chas. R. Vincent & Co., 15 Cortlandt St., New York.

May 14, 1890—SMITH, WALTER W...Pres. Smith & Vaile Co., Dayton, Ohio.

May 21, 1884—SNELL, HENRY I.......... M. E., 135 North Third Street, and 1510 Centennial Ave., Philadelphia, Pa.

May 14, 1890—SNOW, SYLVESTER M....Hd. Dft'sman, W. A. Harris Stm. Eng. Co., 327 Gano St., Prov., R. I.

May 15, 1889—SNOW, WM. W.......Supt. and Gen. Mgr. Ramapo Wheel and Foundry Co., Hillburn, N. Y.

Nov. 30, 1886—SORGE, A., JR.....Mech. Eng. and Exp't, Bee Hive Bldg., Graves St., Rochester, N. Y.

May 14, 1890—SORNBORGER, EDWIN C....Engineer Gordon Steam Pump Co., Hamilton, O., and 180 N. Division St., Buffalo, N. Y.

May 21, 1884—SORZANO, JULIO FEDERICO (*Life Member*)....Cons. Civ. Eng., 33 Broadway, Box 2675, New York City.

April 7, 1880—SOULE, RICHARD H..Assist. Gen. Mgr. Union Sw. & Sig. Co., Swissvale, Pa.

Nov. 29, 1887—SOUTHWORTH, EDWARD P. B..............Mech. D'tsman Penna. Steel Co., Steelton, Pa.

Oct. 16. 1888—SOWTER, ISAAC GEORGE.....Mech. Eng. & Supt. Dry Dock Eng. Wks., 48 Orleans St., and 867 E. Congress St., Detroit, Mich.

Oct. 16, 1888—SPANGLER, H. W..P. A. E. U. S. N., Asst. Prof. Dyn. Eng. Univ. of Penn., Philadelphia, Pa.

May 21, 1884—SPIERS, JAMES...........Mech. Eng'r Fulton Iron Works, San Francisco, Cal.

May 21, 1884—SPIES, ALBERT....Mech. Eng., 22 Cortlandt St., New York City, and 901 Summit Ave., J. C. Heights, Jersey City, N. J.

May 14, 1890—SPILSBURY, EDMUND GYBBON ...Managing Director Cooper, Hewitt & Co., Trenton, N. J.

Nov. 29, 1887—SPRAGUE, WM. T........... M'n'g'r Link Belt Mach. Co., 241 Fourth Ave., So. Minneapolis, Minn.

Nov. 11, 1885—SPRAGUE, WILLIAM W...........Gen. Frman. Loco. Dept. C., R. I. & P. Ry. Co., and 5317 Wentworth Avenue, Chicago, Ill.

May 21, 1884—SPRINGER, J. H., SR..............Supt. Niles Tool Works, Hamilton, Ohio.

April 19, 1882—STAHL, ALBERT W..... Asst. Naval Constructor, U. S. N., Union Iron Works, San Francisco, Cal.

* Manager, 1883-86 ; President of the Society, 1889-90.

May 15, 1889—STANWOOD, JAMES B.......Mech. Eng. and Director Tech. School, 22 Carlisle Building, Cincinnati, and Auburn and Evans Sts., Mt. Auburn, Ohio.

May 21, 1884—STARBUCK, GEO. H.....U. S. Supervising Inspector Steam Vessels, Room 152 Post Office Building, New York City.

April 7, 1880—STEARNS, ALBERT.. Supt. Church & Co.'s Chemical Works, 596 Lorimer Street, Brooklyn, N. Y.

June 13, 1883—STEARNS, THOMAS B............Mining and Mech'l Eng'r, 4 Duff Block, Denver, Colorado.

May 31, 1887—STEEL, CHARLES......Mgr. New Home Sewing Mach. Co., 30 Union Square, New York City.

April 7, 1880—STETSON, GEORGE R...........Prest. New Bedford Gas Co., and 55 Bedford St., New Bedford, Mass.

May 8, 1888—STEVENSON, ARCHY A...............Standard Steel Wks., Lewistown, and 220 So. Fourth St., Philadelphia, Pa.

Nov. 30, 1886—STEWARD, JOHN F...............Mechan. and Pat. Expt., Wm. Deering & Co., 1135 Dunning Street, Chicago, Ill.

May 31, 1887—STEWART, WALTER GREENWAY..Mech. Eng. & Pat. Atty., 530 Washington St., and 116 So. Fifth St., Reading, Pa.

Nov. 6, 1884—STILES, NORMAN C........Treas. Stiles & Parker Press Co. Middletown, Conn.

May 31, 1887—STILLMAN, FRANCIS HILL..210 E. 43d St., New York City, and 70 Division Ave., Brooklyn, N. Y.

June 13, 1883—STILLMAN, OSCAR B..................58 William Street, New York City.

April 7, 1880—STIRLING, ALLAN *..........International Boiler Co., 74 Cortlandt St., New York City.

May 14, 1890—STOCKHAM, WM. H....Supt. Ill. Mall. I. Co., 581 Duversey St., and 495 Racine Ave., Chicago, Ill.

April 7, 1880—STONE, HENRY B............Prest. Chicago and Central Union Telephone Cos., Telephone Bldg., 203 Washington St., and 45 Bellevue Place, Chicago, Ill.

May 21, 1884—STONE, JOSEPH......................85 Milk St., Boston, Mass.

May 8, 1888—STRALE, ALLAN......Mech. Eng'r, Fraser & Chalmers, and 207 Park Ave., Chicago, Ill.

June 13, 1883—STRATTON, E. PLATT....Marine & Consult. Eng., Surveyor I. & S. Hulls, for "Record of Amer. and For. Shipp'g," 37 William St., New York City, and College Point, Long Island, N. Y.

April 7, 1880—STRONG, GEORGE S....Chf. Eng. Strong Loco. Co., 45 Broadway, New York City.

May 21, 1884—STUTZ, SEBASTIAN....Mining & Mech'l Eng'r, Lewis Bldg., P. O. Box 112, Pittsburgh, Pa.

May 21, 1884—SUNSTROM, KARL J..............Consult'g Eng'r, Lindesberg, Sweden.

May 8, 1888—SUPLEE, HENRY H.... Mech. Eng'r, Yale and Towne Mfg. Co., Stamford, Conn.

Oct. 16, 1888—SVENSON, JOHN....Res. Mech. Eng. and Insp. West Superior Iron and Steel Co., 41 Coal Exchange, Scranton, Pa.

Nov. 1, 1883—SWAIN, GEORGE F....Prof. of Civil Eng. Mass. Inst. of Tech., Boston, Mass.

* Manager, 1881-84; Vice-President, 1885-87.

April 7, 1880—SWASEY, AMBROSE........Warner & Swasey, Cleveland, Ohio.
May 27, 1885—SWEENEY, JOHN M.............Mech. Eng., Wheeling, W. Va.
April 7, 1880—SWEET, JOHN E..........Straight Line Engine Co., Syracuse, N. Y.
Nov. 29, 1887—SWINSCOE, CHARLES....M'n'g'r Clinton Wire Cloth Co., Clinton, Mass.

April 7, 1880—TABOR, HARRIS....The Tabor Mfg. Co., 111 Liberty St., New York City, and 120 W. Grand St., Elizabeth, N. J.
May 4, 1881—TALLMAN, FRANK G..........Cons. Mech. Eng'r, Beaver Falls, Pa.
Nov. 1, 1883—TATNALL, JAMES EDWARD....Metallurgist and Engineer, Lock Box 43, and 511 Seneca St., So. Bethlehem, Pa.
May 26, 1886—TAYLOR, FRED W......Gen. Mgr. Mfg. & Investment Co., 15 Broad St., N. Y. City.
Nov. 30, 1886—TAYLOR, J. ARCHIE....M. E. The Pusey & Jones Co., and 912 Madison St., Wilmington, Del.
May 26, 1886—TAYLOR, STEVENSON....V.-P. and Gen. Supt. W. & A. Fletcher Co., 266 West St., and 328 West 57th St., New York City.
Nov. 19, 1889—TAYLOR, WARREN H....Supt. Lock Dept. Yale & Towne M'f'g Co., Stamford, Conn.
May 15, 1889—TERRELL, CHARLES E......... ...Box 287, Waterbury, Conn.
May 8, 1888—THOMAS, CHARLES W.....Pendleton & Thomas, Expt. and Cons. Eng., 121 Nassau St., New York City.
June 20, 1880—THOMAS, EDWARD W....Agt. Tremont and Suffolk Mills, and 12 Butterfield St., Lowell, Mass.
Nov. 1, 1883—THOMAS, JOHN......Gen. Supt. Thomas Iron Co., Hokendauqua, Pa.
June 13, 1883—THOMAS, SAMUEL............................Catasauqua, Pa.
May 31, 1887—THOMSON, JOHN......Mech. Eng'r and Mfr. Water Meters and Pr't'g Mach., 12 Temple Court, New York City., and 190 Carlton Ave., Brooklyn.
April 7, 1880—THOMPSON, CHARLES T....Mech. Eng. I. P. Morris Co., and 2247 Richmond Street, Philadelphia, Pa.
May 26, 1886—THOMPSON, EDGAR B.....Chief D'tsman C. & N. W. R'y, and 77 Warren Ave., Chicago, Ill.
May 21, 1884—THOMPSON, EDWARD P ...M. E. Sol. Pat. & Cons. Elect., 5 Beekman St., New York City, and 618 Salem Ave., Elizabeth, N. J.
May 21, 1884—THOMPSON, ERWIN W....M'g'r Southern Cotton Oil Co., Houston, Texas.
May 21, 1884—THORNE, WM. H....Wm. Sellers & Co., Incorporated, and Gowen Av., Mt. Airy, Phila., Pa.
May 21, 1884—THURMAN, GALEN E....M'g'r Ainslie, Cochran & Co., 920 West Main St., Louisville, Ky.
April 7, 1880—THURSTON, ROBERT H.*...Director Sibley College, Cornell University, Ithaca, N. Y.
Nov. 11, 1885—TILDEN, JAMES A........Eng. and Supt. for Hersey Bros., So. Boston, Mass.
May 31, 1887—TOBEY, GEORGE A.......Supt. U. S. E. L. Co., Newark, N. J.

* President, 1880-82.

May 8, 1888—TOLMAN, EDWARD F....Treas. Wheelock Eng. Co., and 18 Catharine St, Worcester, Mass.

May 31, 1887 } TOMPKINS, S......Prof. Prac. Mech. and Supt. Shops.
May 14, 1890 } Univ. of Tenn., Knoxville, Tenn.

May 14, 1890—TORREY, HERBERT GRAY (*Life Member*)..U. S. Assayer, U. S. Assay Office, 30 Wall St., N. Y., and Sterling, N. J.

Nov. 29, 1887—TOWLE, WM. MASON.....Instr. Mech'l Eng'g, and Fmn. Mach. Shop, Sibley Coll., Cornell Univ., and East Hill House, Ithaca, N. Y.

May 14, 1890—TOWL, FORREST M...Assist. Eng. Nat'l Transit Co., Room 81, 26 B'way, N. Y., and 313 President St., B'klyn.

Nov. 2, 1882—TOWNE, HENRY R.*..Pres. Yale & Towne Mfg. Co., Stamford, Conn.

Nov. 2, 1882—TOWNSEND, DAVID ...M. E. Bush Hill Iron Works, and 1723 Wallace St., Philadelphia, Pa.

May 21, 1884—TRAUTWEIN, ALFRED P....Supt. The Hendrick Mfg. Co., Lock Box 686, Carbondale, Pa.

May 8, 1888—TREGELLES, HENRY.....Westinghouse Air-Brake Co., Pittsburgh, Pa.

Nov. 19, 1889—TREMAINE, E. G......Assist. Supt. Mach. Dept. P. Lorillard & Co., 111 First St., and 84 Wayne St., Jersey City, N. J.

May 15, 1889—TRIBE, JAMES ....Chf. D'ftsman Corliss Eng. Co., and 309 Charles St., Providence, R. I.

Nov. 19, 1889—TRILLY, JOSEPH....Chf. Eng. U. S. N. Navy Yard, Portsmouth, N. H.

April 7, 1880—TROWBRIDGE, WM. P.†..Prof. Engineering Columbia College, 49th St. & 4th Ave., New York City.

May 8, 1888—TRUMP, CHARLES N.....Mech. Eng'r, 1311 West 13th St., Wilmington, Del.

Nov. 5, 1880 } TRUMP, EDWARD N...Mech. Eng. The Solvay Process Co.,
May 14, 1890 } Syracuse, N. Y.

May 31, 1887—TUTTLE, EDGAR G....Supt. Alamo and Coahuila Coal Cos., Box 109, Eagle Pass, Tex., and San Felipe, Coahuila, Mex.

May 14, 1890—TYNAN, J. W....W. Broad and Indian Sts., and 83 Whittaker St., Savannah, Ga.

Nov. 1, 1883—UEHLING, EDWARD A...Supt. Blast Furn., Sloss Iron & S. Co., Birmingham, Ala.

Nov. 30, 1886—ULMANN, CHAS. J.......M. E., Bernhard, Ulmann & Co., 113 Grand St., New York City.

April 19, 1882—UNDERWOOD, F. H...Pres. Underwood Mfg. Co., Tolland, Conn.

Nov. 30, 1886—UNGER, JOHN S....Eng'r Western Gas Constr. Co., Fort Wayne, Ind.

May 8, 1888—UNZICKER, HERMAN....Mech. Eng. Gail, Bumiller & Unzicker, Chicago I. W., Willow St. & Hawthorn Ave., Chicago, Ill.

May 21, 1884—UPSON, LYMAN A....Supt. Hartford Carpet Co., Thompsonville, Conn.

May 8, 1888—VAILE, J. HENRY....V.-P. and Supt. The Smith & Vaile Co., Dayton, Ohio.

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* President, 1888–89; Vice-President, 1884–86.

† Manager, 1880–81; Vice-President, 1881–83.

April 7, 1880— } VANDERBILT, AARON....Supt. N. Y. & Cuba M. S. S. Co.,
May 21, 1884— } 113 Wall Street, New York City.
May 14, 1890—VAN ATTA, HARRY..........Mech. Eng. & Assist. Supt. Rathbone, Sard & Co., 239 N. Pearl St., Albany, N. Y.
May 31, 1887—VAN VLECK, FRANK....Chf. Eng'r San Diego Cable Ry. Co., San Diego, Cal.
May 21, 1884—VAN WINKLE, FRANKLIN....Cons. Eng., 91 Liberty Street, New York City.
May 15, 1889—VERASTEGUI, ALBERT.....Mech. Eng., 116½ Havana St., Havana, Cuba.
Nov. 19, 1889—VERNON, WILLIAM G....Goodell & Waters, 3103 Chestnut St., Philadelphia, Pa.
Nov. 19, 1889—VICTÓRIN, ANTHONY ...Mech. and Cons. Eng. Ordnance Dept., Watervliet Arsenal, W. Troy, and 143 First St., Troy, N. Y.
May 21, 1884—VOGT, AXEL S......Mech. Eng., Penn. R. R. Co., Altoona, Pa.
May 15, 1889—VOORHEES, PHILIP R....Couns.-at-Law, 32 Nassau St., and 9 East 10th St., New York City.
Nov. 19, 1889—VOSS, WILLIAM ...Ass't Eng. Fox Solid Pressed Steel Co., 1004 "Rookery," Chicago, Ill.

Nov. 19, 1889—WAGNER, JOHN R ....Ass't to E. B. Coxe, Drifton, Luzerne Co., Pa.
Nov. 2, 1882—WALKER, JOHN....Manager Walker Manufacturing Co., Cleveland, Ohio.
May 21, 1884—WALL, EDWARD B....Supt. M. P., P. C. & St. L. R. R., Columbus, Ohio.
April 7, 1880— } WALLIS, JOHN MATHER.Supt. Mo. Power P. W. and B. R. R.,
Nov. 1, 1883— } Philadelphia, Pa.
May 26, 1886—WALLIS, PHILIP....Mast. Mech. C., B. & Q. R. R., Beardstown, Ill.
Nov. 30, 1886—WALWORTH, ARTHUR C....Prest. Walworth Constr. and Supply Co., 134 Congress St., Boston, Mass.
June 20, 1880—WARD, W. E...Russell, Burdsall & Ward, Port Chester, N. Y.
April 7, 1880—WARNER, WORCESTER R......Warner & Swasey, Cleveland, Ohio.
June 20, 1880—WARREN, B. H................169 Falmouth St., Boston, Mass.
May 26, 1886—WARREN, JOHN E....Agt. Paper and Pulp Mills, Cumberland Mills, Me.
May 21, 1884 } —WARRINGTON, JAMES N....Sec'y Vulcan I. W.,
Nov. 11, 1885 } 86 No. Clinton St., Chicago, Ill.
May 31, 1887—WARRINGTON, JESSE....Mech. Supt. Nordyke & Marmon Co., and 289 Union St., Indianapolis, Ind.
May 21, 1884—WATERMAN, JOHN S....Brown & Sharpe, and 153 Orms St., Providence, R. I.
May 26, 1886—WATSON, WM....Sec'y Amer. Acad. Arts & Sci., 107 Marlboro' St., Boston, Mass.
June 13, 1883—WATTS, GEORGE W........Mech. Eng. Niles Tool Wks., 705 Arch St., Phila., Pa.
Nov. 2, 1882—WEBB, JOHN BURKITT..Prof. App. Math. Stevens Inst. Tech., Hoboken, N. J.

Nov. 30, 1886—WEBBER, HENRY, JR....Dept. Coal Mine Mach., Dickson Mfg. Co., Scranton, and Dunmore, Pa.
April 7, 1880—WEBBER, SAMUEL...Hydr. & Dyn. Eng'r, Charlestown, N. H.
April 7, 1880—WEBBER, SAMUEL S..Asst. Mgr. Trenton Iron Co.,Trenton, N. J.
May 4, 1881—WEBBER, WILLIAM OLIVER....Gen. Supt. Erie City I. W., and 335 W. 8th St., Erie, Pa.
June 13, 1883—WEBSTER, HOSEA........Mech. Eng. H. R. Worthington, 95 Lake St., and 58 Woodland Ave., Chicago, Ill.
May 21, 1884—WEBSTER, J. F....Supt. Champion Bar, Knife & Machine Co., Springfield, Ohio.
May 27, 1885—WEBSTER, JOHN H..Supt. Standard Ref. Co., 24 Broad Street, Boston, Mass.
May 15, 1889—WEBSTER, WILLIAM R....C. E. and Bridge Inspector, 424 Walnut St., Phila., Pa.
Aug. 10, 1881—WEEKS, GEORGE W.*....Agent Lancaster Mills, Clinton, Mass.
May 15, 1889—WEICKEL, HENRY....Asst. Foreman Drawing Room Yale & Towne Mfg. Co., and 69 Gardner Street, Stamford, Conn.
April 7, 1880—WEIGHTMAN, WM. H...Mech. Eng. and Patents, Para Bldg., 35 Warren St., New York City, and 519 Halsey St., Brooklyn, N. Y.
Nov. 19, 1889—WELLINGTON, A. M....Editor "Eng'ng News" and Cons. Eng., Tribune B'ld'g, and 34 Gramercy Park, N. Y. City.
May 4, 1881—WELLMAN, SAMUEL T.†.... 1080 Wilson Ave., Cleveland, Ohio.
April 7, 1880—WELLS, EBEN F....Cash'r W. S. Iron and Steel Co., and at W. S. Hotel, West Superior, Wis.
May 27, 1885—WELLS, J. LELAND....Gillis & Geoghegan, 116 Wooster St., and 1238 Washington Ave., New York City.
Nov. 19, 1889—WENDT, ARTHUR F....Cons. Eng. The Compania Huanchaca de Bolivia, 13 Park Row, and Antofagasta, Chile, and 127 E. 91st St., New York City.
May 21, 1884—WEST, THOMAS D....Mgr. Thomas D. West Foundry Co., Cleveland, Ohio.
May 21, 1884—WESTINGHOUSE, H. HERMAN ‡..Gen. Man. Westinghouse Air Brake Co., Liberty Ave. & 25th St., Pittsburgh, Pa.
Nov. 2, 1882—WESTON, EDWARD........Elec. Eng., 645 High St., Newark, N. J.
April 7, 1880—WHEELER, F. MERIAM.......George F. Blake Mfg. Co., and Wheeler Condenser & Eng'g Co., 93 Liberty St., N. Y. City, and Montclair, N. J.
June 13, 1883—WHEELER, HERBERT A..Adj. Prof. of Mining, Washington University, St. Louis, Mo.
Nov. 19, 1889—WHEELER, SCHUYLER S...........Prest. Crocker-Wheeler Perfected Motor Co., 430-432 W. 14th St., New York City.
April 7, 1880—WHEELOCK, JEROME........25 Elizabeth St., Worcester, Mass.
May 21, 1884—WHITAKER, EZRA J....Chief Eng. U. S. N., U. S. Rec'v'g Ship "Vermont," Navy Yard, Brooklyn, N.Y.
April 19, 1882—WHITE, H. C.....................16 Dey St., New York City.
June 20, 1880—WHITE, JOSEPH J........Pres. H. B. Smith Machine Co., New Lisbon, N. J.

* Vice-President, 1889-91. † Manager, 1885-88. ‡ Manager, 1889-92.

April 19, 1882—WHITE, MAUNSEL........Bethlehem Iron Co., Bethlehem, Pa.
Oct. 16, 1888—WHITE, WILLIAM, JR....Mining Eng., 35 Fifth Ave., Pittsburgh, Pa.
May 31, 1887—WHITEHEAD, GEORGE E....Supt. Nut & Bolt W'ks, Rhode Island Tool Co., West River St., and 299 Broadway, Providence, R. I.
May 26, 1886—WHITEHILL, ROBERT..Mfg. Eng's, Boilers and Refrig. Mach., Grand Ave., Newburgh, N. Y.
Nov. 1, 1883—WHITHAM, JAY M....Prof. Engineering Ark. Ind. Univ., Fayetteville, Ark.
April 7, 1880—WHITING, GEORGE B..Mech. Eng'r, Harlan & Hollingsworth, Wilmington, Del.
April 7, 1880—WHITING, S. B *....Gen. Man. Calumet & Hecla Mining Co., 12 Ashburton Place, Boston, and 11 Ware St., Cambridge, Mass.
Nov. 29, 1887—WHITLOCK, ROGER HADDOCK....Prof. Mech. Tex. State Ag'l and Mech. College, College Sta., Brazos Co., Tex.
May 26, 1886—WHITNEY, BAXTER D....................Winchendon, Mass.
Nov. 19, 1889—WHITTEMORE, D. J..........Chf. Eng. C., M. & St. P. Ry., Room 51, Chamber of Commerce, and 222 Biddle St., Milwaukee, Wis.
May 26, 1886—WHITTIER, CHAS....Prest. & Gen'l Mgr., 1176 Tremont St., and 50 Winthrop Street, Boston, Mass.
May 21, 1884—WIGHTMAN, D. A......Supt. Pittsburgh Loco. Works, and 154 Locust St., Allegheny, Pa.
Nov. 19, 1889—WILBRAHAM, THOMAS.....Wilbraham Bros., 2518 Frankford Ave., and Main and Wakeling Sts., Frankford, Phila., Pa.
May 31, 1887—WILCOX, JOHN F..Eng. with J. P. Witherow, Lewis Block, 608, and 6th Ave. and Smithfield St., Pittsburgh, Pa.
April 7, 1880—WILCOX, STEPHEN......The Babcock & Wilcox Co., 30 Cortlandt Street, New York City.
June 20, 1880—WILDER, MOSES G.........Mfr. Volumetric Regulators, 816 Cherry St., Philadelphia, Pa.
June 20, 1880—WILEY, WM. H †............53 East 10th St., New York City.
Oct. 16, 1888—WILKIN, W. M.....Des. & Slsman. Stearns Mfg. Co., Erie, Pa.
May 15, 1889—WILLCOX, CHARLES H....Mech. Eng. Willcox & Gibbs S. M. Co., 658 Broadway, and 32 E. 62d St., N. Y. City.
Nov. 19, 1889—WILLIAMS, J. C... Foreman Mach. Dept. Jarecki Mfg. Co., and 145 E. 12th St., Erie, Pa.
Nov. 2, 1882—WILLIAMSON, WM. C....Williamson Bros. E. & B. Works, Richmond & York Streets, and 1800 N. 16th St., Philadelphia, Pa.
June 13, 1883—WILLSON, FREDERICK NEWTON....Prof. Desc. Geom. and Tech. Drawing, Princeton College, Princeton, N. J.
Nov. 29, 1887—WILSON, JAMES EDWARD....1500 Ninth Ave., and 205 West 122d St., New York City.
May 26, 1886—WINTER, HERMAN.........Const. Eng. & Naval Architect, Pier 11, N. R., New York City, and 165 Taylor St., Brooklyn, N. Y.

* Manager, 1880-82; Vice-President, 1882-83. † Treasurer, 1884-90.

Aug. 10, 1881—WITHEROW, JAMES P........Eng. and Contr., Lewis Block, and 6th Ave. and Smithfield St., Pittsburgh, Pa.

Nov. 19, 1889—WOLF, OTTO C.....Eng'r and Arch., Broad and Arch Sts., and 1706 Master St., Phila., Pa.

April 7, 1880—WOLFF, ALFRED R......Consult. Eng., 315 Potter Building, 88 Park Row, New York City.

May 15, 1889—WOHL, LOUIS.....Mech. Eng. Fraser & Chalmers, and 146 Eugenie St., Chicago, Ill.

May 31, 1887—WOOD, DE VOLSON *....Prof. Eng'g Stevens Inst. Tech., Hoboken, and Boonton, N. J.

May 14, 1890—WOOD, E. J....Cons. Eng. and Contractor, 243 Broadway, N. Y., and 828 Quincy St., B'klyn.

May 14, 1890—WOOD, MATTHEW P..Mech. Eng., Supt. Douglass Axe Mfg. Co., 123 W. 21st St., N. Y., also Supt. Archer Gas Fuel Process Co., Pittsburgh, Pa.

Nov. 19, 1889—WOOD, W. H......M. E. with Bement, Miles & Co., 21st and Callowhill Sts., Philadelphia, and Media, Delaware Co., Pa.

April 7, 1880 }
May 14, 1890 } WOOD, WALTER.... Mgr. for R. D. Wood & Co., 400 Chestnut St., Phila., Pa.

April 7, 1880—WOODBURY, C. J. H.†.....2d V.-P. Man'f'rs' Mut. Ins. Co., 31 Milk Street, P. O. Box 112, Boston, and 61 Commercial St., Lynn, Mass.

June 20, 1880—WOODBURY, L. S.............778 Fourth Ave., Detroit, Mich.

June 13, 1883—WOODS, ARTHUR T....Prof. Mech. Eng., University of Ill., and 308 W. Church St., Champaign, Ill.

May 26, 1886—WOODWARD, CALVIN M....Prof. Math. and App. Mech., Washington Univ., St. Louis, Mo.

May 26, 1886—WOOLSON, OROSCO C........Mgr. Amer. S. B. Ins. Co., 781 Broad St., and 239 Mt. Pleasant Ave., Newark, N. J.

May 15, 1889—WOLCOTT, FRANK P....Supt. Colwell I. W., and Carteret, N.J.

Oct. 16, 1888—WORCESTER, FRANKLIN E..Room 47 G. C. Station, N. Y. City.

Nov. 2, 1882—WORTHINGTON, CHARLES C.‡........H. R. Worthington, 145 Broadway, and 29 Fifth Ave., New York City.

May 14, 1890—WRIGHT, LOUIS S....Supt. Link Belt Eng. Co., Nicetown, and 1804 Wallace St., Phila., Pa.

May 21, 1884—WRIGHT, JOHN Q.... Putnam Machine Co., 115 Liberty St., New York City.

May 26, 1886—WYMAN, HORACE W....Worcester Drop Forging Works, 30 Bradley Street, Worcester, Mass.

May 15, 1889—YORK, L. D........Supt. Burgess Steel and Iron Co., Portsmouth, Ohio.

Nov. 29, 1887—YOST, THOMAS MILTON..Mech. Eng'r Am. Tube and Iron Co., Middletown, Pa.

May 14, 1890—ZEHNDER, CHAS. H....Vice-Prest. and Gen. Mgr. Jackson & Woodin Mfg. Co., Berwick, Penn.

May 21, 1884—ZIMMERMANN WM. F.....Gen. Supt. U. S. E. I. Co., Factory 29 Plane St., Newark, N. J.

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* Vice-President, 1889-91. † Manager, 1882-85; Vice-President, 1887-89. ‡ Manager, 1883-86.

Associates.

Nov. 2, 1882—ALLER, A.....Mech. and Cons. Engineer, 109 Liberty Street, New York City, and Richmond Hill, L. I.

Nov. 29, 1887—BAGG, S. F......Sec'y and Treas. Watertown St'm Eng. Co., Watertown, N. Y.

June 20, 1880—BAILEY, E. B ....Manag'r The E. Horton & Son Co., Windsor Locks, Conn.

May 4, 1881—BAILEY, W. H....Agt. Am. Tube Works, 20 Gold Street, and 913 Seventh Ave., New York City.

Nov. 19, 1889—BALDWIN, WM. H...........Youngstown Rolling Mill Co., and 1161 McGuffey St., Youngstown, Ohio.

May 31, 1887—BROOKS, THOMAS H.....Iron F'der, 708-712 Lake St., and 627 Prospect St., Cleveland, Ohio.

Oct. 16, 1888—BURNHAM, WM....Sec'y and Treas. The Standard Steel W'ks, 220 So. Fourth St., and 4301 Spruce St., Philadelphia, Pa.

Nov. 19, 1889—DARLING, EDWIN..........Supt. Pawtucket Water Works, 140 Broadway, Pawtucket, R. I.

Nov. 30, 1886—DICK, JOHN........Mang'r Phœnix Iron Works, Meadville, Pa.

May 15, 1889—DODGE, WALLACE H....Pres. Dodge Mfg. Co., Mishawaka, Ind.

May 21, 1884—DOUGLAS, E. V.............335 Bullitt Bdg., Philadelphia, Pa.

June 13, 1883—EVANS, EDWIN T..L. S. Transit Co., 189 North St., Buffalo, N. Y.

June 13, 1883—
Jan. 15, 1889— FITCH, CHARLES H.....58 Olive St., New Haven, Conn.

May 21, 1884—GIBSON, WM., JR.......Publisher *Engineering and Building Record*, 227 Pearl Street, New York City.

May 31, 1887—GILKERSON, JAMES A.......Gilkerson Mach. Works, Homer, Cortland Co., N. Y.

Nov. 1, 1883—HALL, JOHN H......Gen'l Manager Colt's Pat. Fire Arms Co., Hartford, Conn.

Oct. 16, 1888—HALLOCK, JOHN KEESE..Att'y-at-Law, 8 So. Park Row, and 328 W. 6th St., Erie, Pa.

Nov. 29, 1887—HASKINS, HARRY S.......Edwin Harrington, Son & Co., 1505 Penn Ave., and 1521 Bouvier St., Philadelphia, Pa.

May 31, 1887—HUSTON, CHARLES L.........Gen. Mgr. Lukens I. & S. Co., Coatesville, Pa.

Nov. 29, 1887—LEMOINE, LOUIS RICE....R. D. Wood & Co., 400 Chestnut St., and Chestnut Hill, Philadelphia, Pa.

May 26, 1886—LOW, FRED R.... .....Mech. Editor *Power*, 113 Liberty St., New York, and 121 Sumner Ave., Brooklyn, N. Y.

May 15, 1889—McCOLLIN, THOMAS H....Photographic Chemist, 635 Arch St., Philadelphia, and Lansdowne, Del. Co., Pa.

Oct. 16, 1888—McFARREN, S. J....Contractor, 111 Water Street, Pittsburgh, Pa., and Walker Flats, McKeesport, Pa.

May 15, 1889—Magee, Frank A..........Salesman Electl. Dept. The E. S. Greely Co., 5 and 7 Dey Street, New York City.
May 14, 1890—March, P. G...Mgr., Sec. & Treas. Universal Radial Drill Co., Third & Eggleston Aves., also Fernbank, Cinn., O.
April 7, 1880—Miller, Horace B....*American Machinist*, 96 Fulton Street, New York City.
April 7, 1880—Moore, Charles A......Manning, Maxwell & Moore, 111 and 113 Liberty St., New York City.
April 7, 1880—Moore, Lycurgus B.*.....*American Machinist*, 96 Fulton Street, New York City.

May 31, 1887—Noye, Albert A........Steam Eng. Dept. Jno. T. Noye Mfg. Co., Buffalo, N. Y.

May 8, 1888—Pierce, Walter L....Sec'y Lidgerwood Mfg. Co., 96 Liberty St., and 158 West 82d St., N. Y. City.
May 14, 1890—Pomeroy, L. R....Spec. R.R. Agt., Carnegie, Phipps & Co., Ltd., 45 B'dway, N. Y.
Nov. 6, 1884—Porter, Geo. A...With Porter, Jackson & Co., So. Chicago, Ill.

May 15, 1889—Raymond, James H......Pat. Attorney, 225 Dearborn Street, Chicago, and Evanston, Ill.
May 14, 1890—Redwood, Iltyd I......Analytical Chemist, Queens Co. Oil Works, 141 Kent St., Brooklyn, N. Y.
Nov. 30, 1886—Rupley, Geo......Gen'l Supt. Union Imp't and Elevator Co., Duluth, Minn.
May 26, 1886—Russell, C. M......Sec'y Russell & Co., Massillon, Ohio, and Pres. Standard Horse Nail Co., New Brighton, Pa.

Oct. 16, 1888—Selden, George...............Pres. Erie City I. W., Erie, Pa.
May 15, 1889—Simpson, George R...1st Asst. Examiner U. S. Patent Office, No. 30 Patent Office, and 714 Eleventh Street, N. W., Washington, D. C.
May 8, 1888—Smith, S. Decatur......Iron Fdry., E. York and Moyer Sts., and 1927 Spruce St., Philadelphia, Pa.
June 20, 1880—Sperry, Charles..Port Washington, Queens County, L. I., N. Y.
May 15, 1889—Stangland, B. F.........Supt. Mech. Ventilation, Howard & Morse, 45 Fulton Street, New York City.
Nov. 19, 1889—Stevens, E. A...Prest. Hoboken Ferry Co., Treas. Hackensack Water Co., 3 Newark St., Hoboken, N. J.
Nov. 1, 1883—Stockly, Geo. W.......Pres't Brush Electric Co., and 700 Euclid Ave., Cleveland, Ohio.

May 31, 1887—Temple, W. C......Pittsburgh Mgr. Babcock & Wilcox Co., Room 408, Lewis Block, and 324 River Ave., Pittsburgh, Pa.
May 15, 1889—Tucker, William B...........Mech. Eng., 225 Broad Street, and 36 Murray, Elizabeth, N. J.

May 15, 1889—Watson, H. F.................. Paper Manufacturer, Erie, Pa.
May 26, 1886—Whitney, Wm. M........................Winchendon, Mass.

* Secretary, April 7, 1880, to Nov. 4, 1880; Treasurer, April 7, 1880, to Dec. 2, 1881.

Juniors.

May 15, 1889—ABORN, GEORGE P....D'ftsman Knowles Steam Pump Works, Warren, Mass.
May 15, 1889—AGUILERA, A., JR.........Cons. Eng'r, Puerto Principe, Cuba.
Nov. 19, 1889—ALBREE, CHESTER B..Ornamental Iron Works, 20 Market St., and 191 Ridge Ave., Allegheny, Pa.
May 14, 1890—ANDERSON, LARZ W..Assist. Supt. Addyston Pipe & Steel Co., Addyston, O., and 75 Pike St., Cinn., O.
May 15, 1889—ANDERSON, W. E.....Adjunct Prof. Eng. and Supt. of Shops, Ark. Ind. Univ., Fayetteville, Ark.
Nov. 30, 1886—ARMSTRONG, E. J.........Straight Line Eng. Co., and 337 Delaware Ave., Syracuse, N. Y.

April 19, 1882—BABCOCK, W. IRVING......Manager Chicago Ship Bldg. Co., 101st St. and Calumet River, So. Chicago, Ill.
May 15, 1889—BANG, HENRY A.............Eng. Nat. Water Purifying Co., 145 Broadway, and 1214 Broadway, N. Y. City.
May 15, 1889—BARR, HARRY P........ Thomson-Houston Electric Co., R. R. Dept., Boston, Mass.
Nov. 19, 1889—BASFORD, GEO. M....D'tsman, C. B. & Q. Ry., and 205 Main St., Aurora, Ill. and Parker Hill Ave., Boston, Mass.
May 14, 1890—BATES, ALBERT H....Student-at-law, 22 Wilshire Bldg., & 715 Case Ave., Cleveland, Ohio.
May 27, 1885—BEACH, CHARLES S.....H. E. Bradford & Co., Lock Box 124, and 910 Main St., Bennington, Vt.
May 15, 1889—BIRD, WILLIAM W ....Inst'r in Stm. Eng., Polytech. Institute, Worcester, Mass.
May 14, 1890—BULLOCK, EDWIN R....Dftsman. & Mech. Eng., Conant Thread Co., & 15 So. Union St., Pawtucket, R. I.
May 8, 1888—BURCHARD, ANSON W....Mech. Eng. J. M. Ives Co., 257 Main St., Box 8, Danbury, Conn.
April 7, 1880—BURDSALL, ELWOOD, JR...Port Chester, Westchester Co., N. Y.
Nov. 29, 1887—BURGESS, ISAAC CHESBRO....Inspector, 31 Milk St., Boston, and 49 Austin St., Hyde Park, Mass.
May 8, 1888—BURNS, A. L........D'tsman, Jabez Burns & Sons, 3 Worth St., New York City, and 297 Halsey St., Brooklyn.

May 14, 1890—CHURCHILL, W. W....Westinghouse, Church, Kerr & Co., 17 Cortlandt St., N. Y.
May 8, 1888—COLE, L. W.M. M. Eagle Square Mfg. Co., So. Shaftsbury, Vt.
May 31, 1887—CONRAD, HUGH VINCENT...Rand Drill Co., 23 Park Place, and N. Tarrytown, New York.
May 14, 1890—COOPER, HENRY R....Supt. Caustic Dept. Solvay Process Co., and 420 Milton Ave., Syracuse, N. Y.
May 8, 1888—CURTIS, RALPH E.......209 Juniata St., Trans. Dept. Penna. Lines, Allegheny, Pa.

May 14, 1890—DASHIELL, BENJ. J., Jr....Cons. and Contracting Engineer, 6 South St., Baltimore, Md.

May 14, 1890—DAWES, ROBT..F'm'n Machinist, Erie & Trenton Aves., and 3632 Frankford Ave., Phila., Pa.
May 21, 1884—DAY, F. M .... Hopedale, Mass.
May 15, 1889—DEWSON, EDWARD H., JR... Dist. Fmn. U. P. Ry., Ellis, Kan.
Nov. 30, 1886—DIXON, W. F.............. Cooke L. & M. Co., Paterson, N. J.
May 14, 1890—DINKLE, Jr., GEO.... Assist. Eng., F. O. Matthiessen & Wiechers, 165 Washington St., Jersey City, N. J.
May 8, 1888—DOCKAM, EDWARD HERBERT ....Asbestos Pckg. Co., 169 Congress St., Boston, Mass.
Nov. 19, 1889—DRAVO, GEO. P....Fraser and Chalmers, and 193 Dearborn Ave., Chicago, Ill.

May 15, 1889—EARLE, EDGAR P.......Ordnance Dept., Bethlehem I. Co., South Bethlehem, Pa.
May 15, 1889—EBERHARDT, F. L'H......(Firm of E. Gould & Eberhardt), and 97 Congress St., Newark, N. J.
May 15, 1889—EDWARDS, V. E....D'tsman, 25 Lincoln St., Worcester, Mass.
May 14, 1890—EILERS, KARL EMRICH...Care of S. Bleichröder, Behren St. 63, Berlin, Germany.

May 8, 1888—FIRESTONE, JOE F...F'm'n Mach. Shop Columbus Buggy Co., and 55 Hoffman Ave.,Columbus, Ohio.
Nov. 29, 1887—FORAN, GEO. J...Geo. F. Blake Mfg. Co., 111 Federal St., Boston, and Sanborn Ave., Dorchester, Mass.
May 27, 1885—FOSTER, ERNEST H.....H. R. Worthington Hydr. Works, 86 Liberty St., N. Y., and Englewood, N. J.

May 31, 1887—GARFIELD, ALEXANDER STANLEY......Box 1641, Boston, Mass.
May 8, 1888—GLASSER, CHARLES, H......................Ironwood, Mich.
Nov. 19, 1889—GORTON, JOHN C........D'tsman, Yale & Towne Mfg. Co., and Park Ave., Stamford, Conn.
May 15, 1889—GOSS, EDWARD O...............D'ftsman and Mech. Eng. Scoville Mfg. Co., Waterbury, Conn.
May 21, 1884—GUTHRIE, EDWARD B.......Deputy City Eng'r, Buffalo, N. Y.

May 26, 1886—HIGGINS, GEO. F....Amoskeag Manuf. Co., Manchester, N. H.
May 8, 1888—HILDRETH, WILLIAM OSGOOD...D'tsman Stanley Mfg. Co., Box 193, and 21 Essex St., Lawrence, Mass.
May 8, 188—HOBART, JAMES C....Sec'y The Triumph Cmpd. Eng. Co., 211-219 W. Second St., and Price Hill, Cincinnati, Ohio.
May 15, 1889—HOLMES, CHARLES L...........Mech. Eng., Waterbury, Conn.
Nov. 29, 1887—HUSON, WINFIELD S......Mech. D'tsman & Des. Campbell P't'g Press & Mfg. Co., and Box 365, Taunton, Mass.

Nov. 19, 1889—JARECKI, ALEX. H..... Supt. F'd'y Jarecki Mfg. Co., Ltd., and French and Ninth Streets, Erie, Pa.
Nov. 19, 1889—JOHN, HARRY P.........Chf. D'ftsman St. L. & S. F. Co., and Ozark Hotel, N. Springfield, Mo.
May 15, 1889—JOHNSTON, EDWARD B..Chief D'ftsman Frey Printing Co., 11 and 13 E. 8th St., Cincinnati, Ohio, and Fairfield Ave., Bellevue, Ky.

May 15, 1889—KIELY, H. J.....Salesman Link Belt Eng. Co., 49 Dey St., N. Y. City, and 113 Nevins St., Brooklyn, N. Y.

May 15, 1889—LAIRD, JOHN A.......Asst. Mech. Eng. W. W. Extension, 77 E. May Street, St. Louis, Mo.
Nov. 30, 1886—LIDGERWOOD, WM. VAIL........... Lidgerwood Mfg. Co. 96 Liberty Street, N. Y. City.
Oct. 16, 1888—LIPPS, HENRY, JR............Eng'r M. W. R. & D. R. R., Raleigh, N. C.
Nov. 19, 1889—LOCKWOOD, J. F............D'ftsman Otis Brothers & Co., 38 Park Row, and 2002 Fifth Ave., New York City.
May 15, 1889—LOWE, WILLIAM V........Care of Parkhill Mfg. Co., and 27 Mechanic St., Fitchburg, Mass.
May 31, 1887—LYALL, WILLIAM L.........J. & W. Lyall, 540 W. 23d St., and 367 W. 20th St., New York City.

May 14, 1890—MAGOUN, HENRY A....Ch. D'tsman, Marine Dept. Penna. Steel Co., Sparrow's Point, Md.
April 7, 1880—MARX, HENRY....Gen Agt., The G. A. Gray Co., 481 Sycamore St., Cinn., O.
Oct. 16, 1888—MCLEOD, HOWARD D.......D'tsman and Asst; Eng'r Tests, E. P. Ellis & Co., and cor. Cass and Mason Streets, Milwaukee, Wis.
Nov. 29, 1887—MERRIAM, HENRY P..........Pneum. Dyn. Gun Co., and 11 Charlton St., New York City.
Oct. 16, 1888—MERRILL, ALLYNE LITCHFIELD.........Instr. Mech. Eng'g Mass. Inst. Tech., Boston, and 65 Dana St., Cambridge, Mass.
Oct. 16, 1888—MILLER, EDWARD FURBER.......Instr. Mech. Eng'g Mass. Inst. Tech., Boston, and 9 Franklin St., Cambridgeport, Mass.
May 26, 1886—MORAN, DANIEL E., with Sooysmith & Co., 2 Nassau St., and 26 West 18th St., New York City.
May 31, 1887—MUMFORD, EDGAR HUIDEKOPER..... Supt. Russel Wheel and Fdy. Co., and 210 McDougal Ave., Detroit, Mich.

April 7, 1880—NEFTEL, KNIGHT...........41 Liberty Street, New York City.
Nov. 11, 1885—NORRIS, R. VAN A..Asst. Eng. P. R. R. Coal Co., Box 726, Wilkesbarre, Pa.

Nov. 5, 1880—OWEN, EDWARD H., JR.....150 Crescent St., Waltham, Mass.

May 15, 1889—PERCIVAL, GEORGE S....Westinghouse, Church, Kerr & Co., 17 Cortlandt St., N. Y., and No. 32 W. 46th St., N. Y. City.
Oct. 16, 1888—POLLEDO, Y..Mgr. Sta. Barbara Sugar Works, Cardenas, Cuba.

May 15, 1889—QUIMBY, WILLIAM E.............John Patten Mfg. Co., Mills Bldg., N. Y., and 31 Oakwood Avenue, Orange, N. J.

May 15, 1889—REIST, H. G.. ...E. E. Thomson-Houston Elect. Co., and 113 Franklin St., Lynn, Mass.

May 15, 1889—REYNOLDS, GEORGE F.... M. C. Bullock Mfg. Co., Supt. Diam. Drill Expl., Johannisberg, S. Africa, and care of G. B. Reynolds, Room 20, *Tribune* Bldg., 832 Hinman Ave., Chicago, also Evanston, Ill.

June 20, 1880—RIDGLEY, WM. BARRETT........V.-P. Springfield Iron Co. Springfield, Ill.

May 14, 1890—ROSS, E. L...M. E. Chapman Valve Co., Indian Orchard, Mass.

Nov. 11, 1885—ROWLAND, CHAS. B..............Continental Iron Works, Brooklyn, N. Y., and 329 Madison Ave., New York City.

Nov. 29, 1887—ROWLAND, GEO...............Continental Iron Works and 329 Madison Ave., New York City.

May 15, 1889—SAMUELS, JONATHAN H..D'ftsman Williams, White & Co., Moline, Ill.

May 15, 1889—SCHOLL, JULIAN S.............. F'dry and Mach'n'y Dept. Harrisburg Mfg. Co., and 319 Hammel St., Harrisburg, Pa.

Oct. 16, 1888—SCHWARZ, FRANZ H..........D'tsman Lower Pacific Mills, and 219 Essex St., Lawrence, Mass.

Nov. 19, 1889—SEARS, WILLARD T....D'ftsman Bement, Miles & Co., and 1030 Green St., Phila., Pa.

May 15, 1889—SHEPARD, FRANK E....Mech. Eng'r, of Thomas, Shepard & Searing, 69 Arapahoe Bldg, Denver, Col.

April 7, 1880—SMITH, ALBERT W................Asst. Prof. Mech. Eng. Cornell University, Ithaca, N. Y.

May 15, 1889—SMITH, C. M. W..........Instr. Brooks Tech. School, and D'ftsman and Des., Dunkirk Eng'g Co., and 423 Central Ave., Dunkirk, N. Y.

Oct. 16, 1888—SMITH, CHARLES P.................D'tsman and Designer C. B. Rogers & Co., and 52 Church St., Norwich, Conn.

May 26, 1886—STEVENS, WM. N......N. Y. Safety Car Htg. & Ltg. Co., 160 B'way, also 97 Pine St., New York City.

May 26, 1886—STONE, WILBUR M.... Asst. Supt. The Pratt & Cady Co., and 6 Townley St., Hartford, Conn.

Nov. 1, 1883—SUTER, GEORGE A...Baker, Smith & Co., South Fifth Ave. and Houston St., New York City, and 206 Marcy Ave., Brooklyn, N. Y.

May 8, 1888—TAYLOR, WILLIAM M........Sec'y Chandler & Taylor Co., 370 West Washington St., Indianapolis, Ind.

May 14, 1890—THOMAS, EDWARD G., of Thomas, Shepard & Searing, Mech. & Elect. Engs., Room 69, Arapahoe Bldg, Denver, Col.

May 27, 1885—TORRANCE, KENNETH......Mech. Eng., H. R. Worthington, 95 Lake St., and 261 Dearborn Ave., Chicago, Ill.

May 15, 1889—TROWBRIDGE, FRANK C. ..D'ftsman, Black & Clawson Co., 103 Basin St., Hamilton, O.

May 27, 1885—VAN DUZEE, HAROLD.........Cons. Eng., 430 Walnut St., Room 31, and 1824 Tioga St., Philadelphia, Pa.

May 8, 1888—VEEDER, CURTIS H... ....Thomson-Houston Electric Co., and 113 Franklin St., Lynn, Mass.

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May 14, 1890—WALDRON, FRED. A.....Dftsman. Yale & Towne Mfg. Co., and Myrtle Ave., Stamford, Conn.
Nov. 19, 1889—WATT, S. P..Mech. Eng., Columbus Mach. Co., Columbus, Ohio.
Nov. 19, 1889—WHALEY, W. B. SMITH..........Charleston Iron Works, East Bay and Pritchard Sts., and 26 Legare, Charleston, S. C.
May 27, 1885—WHITING, CHARLES W......Inspector for E. D. Leavitt, 2 Central Sq., and 11 Ware St., Cambridgeport, Mass.
May 15, 1889—WIGGIN, WM. H.................Supt. G. L. Brownell, 16 Union St., and 190 Lincoln St., Worcester, Mass.
Nov. 29, 1887—WILKES, J. FRANK...Supt. Mecklenburg I. W., Charlotte, N. C.
Nov. 11, 1885—WILLIAMS, HARVEY D....Assist. Prof. Mech. Dr. and Mach. Des. Sibley Coll., Cornell Univ., and 100 Cascadilla Place, Ithaca, N. Y.
Nov. 19, 1889—WORCESTER, VERNOR F......D'ftsman Howe Scale Co., and 36 Pine St., Rutland, Vt.

Deceased.

HENRY R. WORTHINGTON..........Dec. 17, 1880.
THEODORE R. SCOWDEN..........Dec. 31, 1881.
ALEXANDER L. HOLLEY..........Jan. 29, 1882.
ERASTUS W. SMITH..........June 12, 1882.
PETER COOPER, Honorary Member..........April 4, 1883.
JAMES PARK, JR. ....April 21, 1883.
W. K. SEAMAN..........July 2, 1883.
REDMOND J. BROUGH..........July 21, 1883.
C. W. SIEMENS, Honorary Member..........Nov. 20, 1883.
HENRY F. SNYDER..........Nov. 25, 1883.
O. HALLAUER, Honorary Member..........Dec. 5, 1883.
WILLIAM ATWOOD..........Feb. 16, 1884.
WILMER G. CARTWRIGHT..........Feb. 23, 1884.
THEODORE H. RISDON..........May 19, 1884.
ISAAC NEWTON..........Sept. 25, 1884.
J. H. BURNETT..........Jan. 31, 1885.
HORACE LORD..........Feb. 28, 1885.
D. H. HOTCHKISS..........April 29, 1885.
HENRI TRESCA, Honorary Member..........June 24, 1885.
HENRY H. GORRINGE..........July 6, 1885.
WILBUR H. JONES..........July 29, 1885.
FREDERIC E. BUTTERFIELD..........Sept. 5, 1885.
WM. CLEVELAND HICKS..........Oct. 19, 1885.
D. S. HINES..........Nov. 9, 1885.
THEODORE BERGNER..........Jan. 5, 1886.
EMILE F. LOISEAU..........April 30, 1886.
JOHN C. HOADLEY..........Oct. 21, 1886.
HOMER HAMILTON..........Nov. 29, 1886.
JOHN B. ROOT..........Dec. 11, 1886.
BISHOP ARNOLD..........Feb. 16, 1887.
B. F. EMERSON (*Associate*)..........April 5, 1887.

Wm. L. Nicoll . . . . July 2, 1887.
Jackson Bailey (*Associate*) . . . . July 7, 1887.
James Sheriffs . . . . July 18, 1887.
William Wallace Hanscom . . . . Jan. 19, 1888.
Barnabas H. Bartol . . . . Feb. 10, 1888.
H. P. Gregory . . . . July 2, 1888.
Alfred B. Couch . . . . Aug. 2, 1888.
Rudolph Clausius, Honorary Member . . . . Aug. 24, 1888.
William Miller . . . . Sept. 21, 1888.
Daniel N. Jones . . . . Dec. 10, 1888.
Cornelius H. Delamater . . . . Feb. 7, 1889.
John Ericsson . . . . March 8, 1889.
Harvey F. Gaskill . . . . April 30, 1889.
Alex. Hamilton, Jr. . . . . May 31, 1889.
Wm. H. Scranton . . . . June 19, 1889.
Henry Parsons . . . . June 20, 1889.
James Beggs . . . . July 19, 1889.
John Coffin . . . . Sept. 3, 1889.
Wm. R. Jones . . . . Sept. 29, 1889.
Howell Green . . . . Oct. 25, 1889.
Fred B. Rice . . . . Dec. 6, 1889.
Hector C. Havemeyer (*Life Member*) . . . . Dec. 14, 1889.
Chas. A. Ashburner . . . . Dec. 24, 1889.
Horatio Allen, Honorary Member . . . . Dec. 31, 1889.
Gustave-Adolphe Hirn . . . . Jan. 14, 1890.
Chas. D. Smith . . . . Jan. 23, 1890.

AMENDED.

RULES

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

[*Adopted November 5th*, 1884.]

OBJECTS.

ART. 1. The objects of the AMERICAN SOCIETY OF MECHANICAL ENGINEERS are to promote the Arts and Sciences connected with Engineering and Mechanical Construction, by means of meetings for social intercourse and the reading and discussion of professional papers, and to circulate, by means of publication among its members, the information thus obtained.

MEMBERSHIP.

ART. 2. The Society shall consist of Members, Honorary Members, Associates and Juniors.

ART. 3. Mechanical, Civil, Military, Mining, Metallurgical and Naval Engineers and Architects may be candidates for membership in this Society.

ART. 4. To be eligible as a *Member*, the candidate must have been so connected with some of the above-specified professions as to be considered, in the opinion of the Council, competent to take charge of work in his department, either as a designer or constructor, or else he must have been connected with the same as a teacher.

ART. 5. *Honorary Members*, not exceeding twenty-five in number, may be elected. They must be persons of acknowledged professional eminence who have virtually retired from practice.

ART. 6. To be eligible as an *Associate*, the candidate must have such a knowledge of or connection with applied science as qualifies him, in the opinion of the Council, to co-operate with engineers in the advancement of professional knowledge.

ART. 7. To be eligible as a *Junior*, the candidate must have been in the practice of engineering for at least two years, or he must be a graduate of an engineering school.

The term "Junior" applies to the professional experience, and not to the age of the candidate. Juniors may become eligible to membership.

ART. 8. All Members and Associates shall be equally entitled to the privileges of membership. Honorary Members and Juniors shall not be entitled to vote nor to be members of the Council.

ELECTION OF MEMBERS.

ART. 9. Every candidate for admission to the Society, excepting candidates for honorary membership, must be proposed by at least three members, or members and associates, to whom he must be personally known, and he must be seconded by two others. The proposal must be accompanied by a statement in writing by the candidate of the grounds of his application for election, including an account of his professional experience, and an agreement that he will conform to the requirements of membership if elected.

ART. 10. All such applications and proposals must be received and acted upon by the Council at least thirty days before a regular meeting, when the Secretary shall at once mail to each member and associate, in the form of a letter ballot, the names of candidates recommended by the Council for election.

ART. 11. Any member or associate entitled to vote may erase the name of any candidate, and may, at his option, return to the Secretary such ballot enclosed in two envelopes, the inner one to be blank and the outer one endorsed by the voter.

ART. 12. The rejection of any candidate for admission as member, associate, or junior, by *seven* voters, shall defeat the election of said candidate. The rejection of any candidate for admission as honorary member by *three* voters shall defeat the election of said candidate.

ART. 13. The said blank envelopes shall be opened by the Council at any meeting thereof, and the names of the candidates elected shall be announced in the first ensuing meeting of the Society, and also in the first ensuing list of members. The names of candidates not elected shall neither be announced nor recorded in the proceedings

ART. 14.—Candidates for admission as honorary members shall

not be required to present their claims; those making the nominations shall state the grounds therefor, and shall certify that the nominee will accept if elected. The method of election in other respects shall be the same as in case of other candidates.

ART. 15. All persons elected to the Society, excepting honorary members, must subscribe to the rules and pay to the Treasurer the initiation fee before they can receive certificates of membership. If this is not done within six months of notification of election, the election shall be void.

ART. 16. The proposers of any rejected candidate may, within three months after such rejection, lay before the Council written evidence that an error was then made, and if a reconsideration is granted, another ballot shall be ordered, at which thirteen negative votes shall be required to defeat the candidate.

ART. 17. Persons desiring to change the class of their membership shall be proposed in the same form as described for a new applicant.

FEES AND DUES.

ART. 18. The initiation fees of members and associates shall be $15, and their annual dues shall be $10, payable in advance. The initiation fee of juniors shall be $10, and their annual dues $5, payable in advance. A junior, being promoted to full membership, shall pay an additional initiation fee of $5. Any member or associate may become, by the payment of $150 at any one time, a life member or associate, and shall not be liable thereafter to annual dues.

ART. 19. Any member, associate or junior, in arrears may, at the discretion of the Council, be deprived of the receipt of publications, or stricken from the list of members, when in arrears for one year. Such person may be restored to membership by the Council on payment of all arrears, or by re-election after an interval of three years.

OFFICERS.

ART. 20. The affairs of the Society shall be managed by a Council, consisting of a President, six Vice-Presidents, nine Managers and a Treasurer, who shall also be the Trustees of the Society.

All past (Ex) Presidents of the Society, while they retain their membership therein, shall be known as Honorary Councillors, and shall be entitled to receive notices of all meetings of the Council

and may take part in any of its deliberations; they shall be entitled to vote upon all questions except such as affect the legal rights or obligations of the Society or its members.

ART. 21. The members of the Council shall be elected from among the members and associates of the Society at the annual meetings, and shall hold office as follows:

The President and the Treasurer for one year; and no person shall be eligible for immediate re-election as President who shall have held that office for two consecutive years; the Vice-Presidents for two years and the Managers for three years; and no Vice-President or Manager shall be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

ART. 22. A Secretary, who shall be a member of the Society, shall be appointed for one year by a majority of the members of the Council at its first meeting after the annual election, or as soon thereafter as the votes of a majority of the members of the Council can be secured for a candidate. The Secretary may be removed by a vote of twelve members of the Council, at any time after one month's notice has been given him by a majority of its members to show cause why he should not be removed, and he has been heard to that effect. The Secretary may take part in any of the deliberations of the Council, but shall not have a vote therein. His salary shall be fixed for the time he is appointed by a majority vote of the Council.

ART. 23. At each annual meeting, a President, three Vice-Presidents, three Managers and a Treasurer shall be elected, and the term of office of each shall continue until the end of the meeting at which their successors are elected.

ART. 24. The duties of all officers shall be such as usually pertain to their offices or may be delegated to them by the Council or by the Society. The Council may, in its discretion, require bonds to be given by the Treasurer.

ART. 25. The Council may, by vote of a majority of all its members, declare the place of any officer vacant, on his failure for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office. All such vacancies and those occurring by death or resignation shall be filled by the appointment of the Council, and any person so appointed shall hold office for the remainder of the term for which his predecessor was elected or appointed; *provided* that the said appointment shall not render him ineligible at the next annual meeting.

ART. 26. Five members of the Council shall constitute a quorum; but the Council may appoint an Executive Committee, or business may be transacted at a regularly called meeting of the Council, at which less than a quorum is present, subject to the approval of a majority of the Council, subsequently given in writing to the Secretary and recorded by him with the minutes. Absent members of the Council may vote by proxy upon subjects stated in the call for a meeting, said proxy to be deposited with the Secretary.

ART. 27. The President on assuming office shall appoint a Finance Committee and a Publication Committee and a Library Committee of five members each. The appointment of two members of each Committee shall expire at the end of each year. The Secretary shall, *ex officio*, be a member of all three Committees.

ART. 28.—The Finance Committee shall have power to order all ordinary or current expenditures, and shall audit all bills therefor. No bill shall be paid except upon their audit. When special appropriations are ordered by the Society, they shall not take effect until they have been referred to the Council and Finance Committee in conference.

ART. 29. It shall be the duty of the Publication Committee to receive all papers contributed, to decide which shall be published in the *Transactions*, and which shall be read in full at the meetings.

ART. 30. It shall be the duty of the Library Committee to take charge of the collection of all material for the Library of the Society, and to supervise all regulations for its use.

ELECTION OF OFFICERS.

ART. 31. At the regular meeting preceding the annual meeting a nominating committee of five members, not officers of the Society, shall be appointed, and this committee shall, at least thirty days before the annual meeting, send to the Secretary the names of nominees for the offices falling vacant under the rules. In addition to such regularly appointed committee, any other five members or associates, not in arrears, may constitute an independent nominating committee, and may present to the Secretary, at least thirty days before the annual meeting, all the names of such candidates as they may select. All the names of such independent nominees shall be placed upon the ballot list with nothing to distinguish them from the nominees of the regular committee, and the Secretary shall at once mail the said list of names to each member and associate in the form of a letter ballot, it being un-

derstood that the assent of the nominees shall have been secured in all cases.

ART. 32. In the election of Vice-Presidents, each member and associate may cast as many votes as there are Vice-Presidents to be elected. He may give all these votes to one candidate, or distribute them among more, as he chooses. Managers shall be voted for in the same way.

ART. 33. Any member or associate entitled to vote may vote by retaining or changing the names on said list, leaving names not exceeding in number the officers to be elected, and returning the list to the Secretary—such ballot inclosed in two envelopes, the inner one to be blank and the outer one to be indorsed by the voter. No member or associate in arrears since the last annual meeting shall be allowed to vote until said arrears shall have been paid.

ART. 34. The said blank envelopes shall be opened by tellers at the annual meeting, and the person who shall have received the greatest number of votes for the several offices shall be declared elected.

MEETINGS.

ART. 35. The annual meeting of the Society shall be held on the first Thursday in November of each year, in the City of New York, unless otherwise ordered, at which a report of proceedings and an abstract of the accounts shall be furnished by the Council. The Council may change the place of the annual meeting, and shall, in that case, give timely notice to members and associates.

ART. 36. Other regular meetings of the Society shall be held in each year at such time and place as the Council may appoint. At least thirty days' notice of all meetings shall be mailed by the Secretary to members, honorary members, associates and juniors.

ART. 37. Special meetings may be called whenever the council may see fit; and the Secretary shall call a special meeting at the written request of twenty or more members. The notices for special meetings shall state the business to be transacted, and no other shall be entertained.

ART. 38. Any member, honorary member or associate may introduce a stranger to any meeting; but the latter shall not take part in the proceedings without the consent of the meeting.

ART. 39. Every question which shall come before the Society shall be decided, unless otherwise provided by these rules, by the votes of a majority of the members and associates present, provided there is a quorum.

Art. 40. At any regular meeting of the Society thirteen or more members and associates shall constitute a quorum.

Art. 41. Unless otherwise ordered, papers shall be read in the order in which their text is received by the Secretary. Before any paper appears in the *Transactions* of the Society a copy of the paper shall be sent to the author, and, so far as possible, a copy of the reported discussion shall be sent to every member who took part in the same, with requests that attention shall be called to any errors therein.

Art. 42. The Society shall claim no exclusive copyright in papers read at its meetings, nor in reports of discussions, except in the matter of official publication with the Society's imprint, as its *Transactions*. The Secretary shall have sole possession of papers between the time of their acceptance by the Publication Committee and their reading, together with the drawings illustrating the same; and at the time of such reading, or as soon thereafter as practicable, he shall cause to be printed, with the authors' consent, copies of such papers, "subject to revision," with such illustrations as are needed for the *Transactions*, for distribution to the members and for the use of technical newspapers, American and foreign, which may desire to reprint them in whole or in part. The policy of the Society in this matter shall be to give papers read before it the widest circulation possible, with the view of making the work of the Society known, encouraging mechanical progress, and extending the professional reputation of its members.

Art. 43. The author of each paper read before the Society shall be entitled to twelve copies, if printed, for his own use, and all members shall have the right to order any number of reprints of papers at a cost to cover paper and printing; *provided*, that said copies are not intended for sale.

Art. 44. The Society is not, as a body, responsible for the statements of fact or opinion advanced in papers or discussions, at its meetings; and it is understood that papers and discussions should not include matters relating to politics or purely to trade.

AMENDMENTS.

Art. 45. These rules may be amended, at any annual meeting, by a two-thirds vote of the members present; *provided*, that written notice of the proposed amendment shall have been given at a previous meeting.

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LIST OF ILLUSTRATIONS.

PAPERS

OF THE

NEW YORK MEETING

(XXth)

NOVEMBER 18th to 23d, 1889.

BEING ALSO THE TENTH ANNUAL MEETING OF THE SOCIETY.

CCCLVII.

PROCEEDINGS

OF THE

NEW YORK MEETING

(XXth)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

November 18th to 23d, 1889.

LOCAL COMMITTEE OF ARRANGEMENTS:—S. W. Baldwin, Chas. Kirchhoff, Jr., J. F. Lewis, J. F. Holloway, Wm. H. Wiley, F. R. Hutton.

FIRST DAY. MONDAY, NOVEMBER 18TH.

The XXth Meeting of the American Society of Mechanical Engineers was also its Tenth Annual Meeting, and was opened by a social reception, held in the rooms of the Society, No. 64 Madison Avenue, beginning at 8.30 on Monday evening. A committee of ladies acted as hostesses to receive the members and their ladies; and later in the evening a collation was served in the rooms above. This reunion was the first of its kind, the Society having only occupied its rooms for a short time. The rooms had been newly furnished, and the walls were hung with pictures, among which was a crayon portrait of Alex. L. Holley, one of the founders of the Society. Over three hundred persons were present, of whom more than fifty were ladies.

SECOND DAY. TUESDAY, NOV. 19TH.

The first session for business was convened in the Hall of the New York Academy of Medicine, No. 12 West 31st St., at ten

o'clock. The secretary's registers showed the following members in attendance during the Convention:

Alberger, Louis R.	New York City.
Alden, Geo. I.	Worcester, Mass.
Allen, Francis B	Hartford, Conn.
Allison, Robert	Port Carbon, Pa.
Almond, T. R.	Brooklyn, N. Y.
Ashworth, Daniel	Pittsburgh, Pa.
Babcock, Geo. H. (*Past President*)	New York City.
Backstrom, G. L	Philadelphia, Pa.
Baker, W. S. G. (*Vice-President*)	Baltimore, Md.
Baldwin, S. W. (*Manager*)	New York City.
Baldwin, W. J	New York City.
Ball, F. H. (*Manager*)	Erie, Pa.
Barnard, Geo. A.	New York City.
Barnes, Abel T.	Jamaica Plain, Mass.
Barnes, D. L.	Chicago, Ill.
Barnhurst, H. R.	Erie, Pa.
Bassett, N. C	Yonkers, N. Y.
Bates, Alex. B.	New York City.
Bayles, Jas. C.	New York City.
Beach, C. S.	Bennington, Vt.
Billings, C. E.	Hartford, Conn.
Binsse, H. L.	Newark, N. J.
Bird, W. W	Worcester, Mass.
Bond, Geo. M. (*Manager*)	Hartford, Conn.
Booraem, J. V. V.	Brooklyn, N. Y.
Borden, T. J. (*Vice-President*)	Fall River, Mass.
Boyd, Jno. T.	Philadelphia, Pa.
Brady, Jas.	Brooklyn, N. Y.
Brooks, Morgan	St. Paul, Minn.
Bulkley, H. W.	New York City.
Burdsall, Elwood, Jr.	Portchester, N. Y.
Burns, A. L.	New York City.
Burpee, G. H.	
Butterworth, Jas.	Philadelphia, Pa.
Caldwell, A. J	New York City.
Campbell, A. C.	Waterbury, Conn.
Cartwright, Robt.	Rochester, N. Y.
Cary, A. A.	New York City.
Cheney, W. L.	Hartford, Conn.
Christensen, A. C.	New York City.
Christiansen, A.	West Troy, N. Y.
Christie, W. W	Hillburn, N. Y.
Clark, W. L.	New York City.
Clarke, S. J.	New York City.
Cloud, Jno. W	Buffalo, N. Y.
Cole, L. W	So. Shaftsbury, Vt.
Coleman, I. B.	Elmira, N. Y.
Colwell, A. W.	New York City.

Conant, T. P.	New York City.
Conrad, H. V.	New York City.
Corbett, C. H.	Brooklyn, N. Y.
Cornelius, H. J.	Philadelphia, Pa.
Crane, T. S.	Newark, N. Y.
Crane, W. E.	Waterbury, Conn.
Cremer, J. M.	Brooklyn, N. Y.
Cruikshank, Barton	Brooklyn, N. Y.
Cullingworth, Geo. R.	New York City.
Dallett, W. P.	Philadelphia, Pa.
Darling, Edwin	Pawtucket, R. I.
Davis, I. H.	Dorchester, Mass
Denton, J. E.	Hoboken, N. J.
Dick, Jno	Meadville, Pa.
Dickey, W. D.	New York City.
Doran, W. S.	New York City.
Drown, F. E.	Pawtucket, R. I.
Dutton, C. S.	Youngstown, Ohio.
Eberhardt, F. L.	Newark, N. J.
Edson, J. B.	New York City.
Emery, A. H.	Stamford, Conn.
Engel, L. G.	Brooklyn, N. Y.
Faber du Faur, A.	New York City.
Falkenau, A.	Philadelphia, Pa.
Field, C. J.	Brooklyn, N. Y.
Fladd, F. C.	New York City.
Fletcher, Andrew	New York City.
Fletcher, W. H.	New York City.
Forney, M. N.	New York City.
Fowler, Geo. L.	New York City.
Freeman, J. R.	Boston, Mass.
Gantt, H. L.	Philadelphia, Pa.
Geohegan, S. J.	New York City.
Gilkerson, J. A.	New York City.
Gilmore, R. J.	Providence, R. I.
Gobeille, J. L.	Cleveland, Ohio.
Gold, S. J.	Englewood, N. J.
Goss, W. O.	Waterbury, Conn.
Goubert, A. A.	New York City.
Gould, W. V.	Norwich, Conn.
Grinnell, F. (*Manager*)	Providence, R. I.
Hague, C. A.	New York City.
Hallock, J. K.	Erie, Pa.
Harmon, O. S.	Brooklyn, N. Y.
Hawkins, J. T. (*Manager*)	Taunton, Mass.
Hayward, F. H.	New York City.
Hazard, V. G.	Wilmington, Del.
Hemenway, F. F.	New York City.
Heggem, C. O.	
Henney, Jno., Jr.	New Haven, Conn.
Henthorn, J. T.	Providence, R. I.

Hewitt, Wm. . . . . Trenton, N. J.
Hill, Wm. . . . . Collinsville, Conn.
Hillman, Gustav. . . . . City Island, N. Y.
Hollingsworth, Sumner. . . . . Boston, Mass.
Holloway, J. F. (*Past President*). . . . . New York City.
Hornig, J. L. . . . . New York City.
Hunt, C. W. . . . . New York City.
Hunt, R. W. . . . . Chicago, Ill.
Hutton, F. R. (*Secretary*) . . . . New York City.
Hyde, C. E. . . . . Bath, Me.
Idell, Frank E. . . . . New York City.
Jacobi, A. W. . . . . New York City.
Jacobus, D. S. . . . . Hoboken, N. J.
Jenkins, W. R. . . . . Bellefonte, Pa.
Jenks, W. H. . . . . Brookville, Pa.
Jones, W. C. . . . . Cincinnati, Ohio.
Kent, Wm. (*Vice-President*). . . . . New York City.
Kirchhoff, Chas., Jr. . . . . New York City.
Krause, Arthur. . . . . Jersey City, N. J.
Laforge, F. H. . . . . Waterbury, Conn.
Lambert, W. C. . . . . New Haven, Conn.
Lane, H. M. . . . . Providence, R. I.
Laureau, L. G. . . . . Philadelphia, Pa.
Leavitt, F. M. . . . . Brooklyn, N. Y.
Lemoine, L. R. . . . . Philadelphia, Pa.
Lewis, J. F. . . . . New York City.
Le Van, W. B. . . . . Philadelphia, Pa.
Lockwood, J. F. . . . .
Low, F. R. . . . . New York City.
Lyall, W. L. . . . . New York City.
Lyne, L. F. . . . . New York City.
MacBride, Jas. . . . . Brooklyn, N. Y.
MacElroy, Samuel. . . . . Brooklyn, N. Y.
MacRae, J. D. . . . . Baldwinsville, N. Y.
MacKinney, W. C. . . . . Philadelphia, Pa.
Mattice, Asa M. . . . . Cambridgeport, Mass.
Meyer, J. G. A. . . . . New York City.
Miller, Alex. . . . . New York City.
Miller, Lebbeus B. . . . . Elizabeth, N. J.
Miller, I. S. . . . . New York City.
Miller, Horace S. . . . . New York City.
Minot, H. P. . . . . Columbus, Ohio.
Montgomery, H. M. . . . . Norwood, Mass.
Morgan, C. H. . . . . Worcester, Mass.
Moore, Lycurgus B. . . . . New York City.
Morris, H. G. . . . . Philadelphia, Pa.
Morse, C. M. . . . . Buffalo, N. Y.
Muller, Maurice A. . . . . Newark, N. J.
Nason, C. W. . . . . New York City.
Naylor, E. W. . . . . New York City.
Nicholson, D. K. . . . . Steelton, Pa.

Nicoll, C. H.
Odell, W. H.................................Yonkers, N. Y.
Parks, E. H.................................Providence, R. I.
Parsons, H. de B.................................New York City.
Partridge, W. E.................................New York City.
Peabody, C. H.................................Boston, Mass.
Percival, G. S.................................New York City.
Perry, W. A.................................New York City.
Pierce, W. L.................................New York City.
Porter, H. F. J.................................New York City.
Pusey, C. W.................................Wilmington, Del.
Ramsay, H. A.................................Baltimore, Md.
Raynal, A. H.................................Richmond, Va.
Robinson, J. M.................................New York City.
Roelker, H. R.................................New York City.
Rogers, W. S.................................West Troy, N. Y.
Russell, W. S.................................Detroit, Mich.
Russell, C. M.................................Massillon, Ohio.
Ruth, W. M.................................Fort Wayne, Ind.
Rowland, C. B.................................Brooklyn, N. Y.
Rowland, George.................................Brooklyn, N. Y.
Schwanhausser, A. W.................................Brooklyn, N. Y.
Scribner, C. W.................................Ames, Iowa.
See, Horace (*Past President*).................................New York City.
Sellers, Coleman (*Past President*).................................Philadelphia, Pa.
Shaw, T. J.................................Wilmington, Del.
Sheldon, T. C.................................Boylston, Mass.
Sinclair, Geo. M.................................Bethlehem, Pa.
Simpkin, Wm.................................Richmond, Va.
Simpson, Geo. R.................................Washington, D. C.
Skinner, L. G.................................Erie, Pa.
Smith, A. P.................................Washington, D. C.
Smith, C. H. L.................................Brooklyn, N. Y.
Smith, Geo. H.................................Providence, R. I.
Smith, Jesse M.................................Detroit, Mich.
Smith, Oberlin.................................Bridgeton, N. J.
Smith, Scott A.................................Providence, R. I.
Smith, Sydney L.................................Boston, Mass.
Smith, T. Carpenter.................................Philadelphia, Pa.
Snell, Henry I.................................Philadelphia, Pa.
Snow, Wm. W.................................Hillburn, N. Y.
Spangler, H. W.................................Philadelphia, Pa.
Sperry, Chas.................................Port Washington, N. Y.
Stangland, B. F.................................New York City.
Stearns, Albert.................................Brooklyn, N. Y.
Steel, Chas.................................New York City.
Stetson, Geo. R.................................New Bedford, Mass.
Stirling, Allan.................................New York City.
Stratton, E. P.................................New York City.
Sunstrom, K. J.................................Worcester, Mass.
Suplee, H. H.................................Philadelphia, Pa.

Swasey, Ambrose............................Cleveland, Ohio.
Sweet, John E. (*Past President*)...............Syracuse, N. Y.
Tabor, Harris..............................New York City.
Taylor, Stevenson..........................New York City.
Terrell, C. E..............................Waterbury, Conn.
Thomas, C. W...............................Newark, N. J.
Thomson, John..............................New York City.
Thompson, E. P.............................New York City.
Thurston, R. H. (*Past President*)..............Ithaca, N. Y.
Torrance, Kenneth..........................Brooklyn, N. Y.
Towne, Henry R. (*President*)..................Stamford, Conn.
Trautwein, A. P............................Carbondale, Pa.
Trowbridge, W. P...........................New York City.
Tucker, W. B...............................Newark, N. J.
Uehling, E. A..............................Bethlehem, Pa.
Ulmann, C. J...............................New York City.
Upson, L. A................................Thompsonville, Conn.
Victorin, Anthony..........................West Troy, N. Y.
Voorhees, P. R.............................New York City.
Ward, W. E.................................Portchester, N. Y.
Warner, W. R...............................Cleveland, O.
Warren, B. H...............................Boston, Mass.
Webb, J. B.................................Hoboken, N. J.
Webber, S. S...............................Trenton, N. J.
Webster, J. H..............................Boston, Mass.
Weeks, Geo. W..............................Clinton, Mass.
Weightman, W. H............................New York City.
Wellman, S. T..............................Cleveland, O.
Wheeler, F. M..............................New York City.
Wheelock, Jerome...........................Worcester, Mass.
Wheeler, S. S..............................New York City.
Weickel, Henry.............................Stamford, Conn.
White, Joseph J............................Philadelphia, Pa.
Whitehead, Geo. E..........................Providence, R. I.
Whitney, B. D..............................Winchendon, Mass.
Whitney, W. M..............................Winchendon, Mass.
Wiley, Wm. H. (*Treasurer*)....................New York City.
Wilkin, W. M...............................Erie, Pa.
Willcox, C. H..............................New York City.
Williamson, W. C...........................Philadelphia, Pa.
Wilson, Jas. E.............................New York City.
Wolff, A. R................................New York City.
Wood, DeVolson.............................Hoboken, N. J.
Woodbury, C. J. H. (*Vice-President*)..........Boston, Mass.
Woolson, O. C..............................Newark, N. J.
Worthington, C. C..........................New York City.
Wyman, H. W................................Worcester, Mass.
Total.............................249.

There was also a number of guests present at the sessions, and

a large number of ladies—over fifty—was present at the reunions, and on the excursions of the meeting.

The retiring president, Mr. H. R. Towne, of Stamford, Conn., delivered his annual address in opening the meeting, and at its close called the docket of routine business. The first order was:

THE ANNUAL REPORT OF THE COUNCIL.

The Council would beg leave to present its Annual Report, as follows:

It has held seven meetings during the year for the transaction of business, and the following is a summary of its action, in addition to the usual routine labor. A report from the Council in reference to the invitation of the Engineers of Great Britain was given at the nineteenth meeting, in Erie, May, 1889. The trip to England has occurred since that report, and the Council has directed that a brief account of the courtesies,—social and professional,—enjoyed by the members of the Society, should be published as an appendix to the Volume of Transactions.

The Society has moved during the year from its former quarters in the Stewart Building, No. 280 Broadway, and now occupies the ground floor of No. 64 Madison Avenue.

The Council has directed that the Society's library should be opened during the evenings until further notice, under the care of the Library Committee.

The co-operation of this Society has been requested in the matter of extending courtesies to the Iron and Steel Institute of Great Britain, which expects to hold a session in this country at some future date in 1890, hereafter to be designated.

The resolutions in connection with the death of Capt. Ericsson have been published in the 10th Volume.

The Council has also directed that the Volumes 1, 2, and 3 of the Transactions, issued in the early history of the Society, and now becoming scarce, should be disposed of only to members of the Society, and at the tariff of $10 per volume, and that an effort be made to ascertain whether there is a sufficient demand for those earlier volumes to justify a second edition of them.

The unprecedented growth of the Society during the last year, and the increase of the work in the Society's office by rea-

son of that growth, has made necessary the employment of additional service, for which arrangements have been made.

The Council has passed favorably upon the applications of one hundred and seventy-eight candidates during the Society's year. The present membership of the Society, including those joining at this meeting, is 1049, distributed among the grades as follows:

Honorary members	17
Life members	9
Members	878
Associates	48
Juniors	97
Total	1049

The Council has directed that the Society Catalogue should contain not only those in active membership, but, also, in an appended list, the names of those whose membership has terminated during the year.*

* It has been thought that it would not be without interest if at the Annual Meeting which completes the first decade of the Society's history, there should be a brief summary noted of the growth of the Society since 1880.

The total membership in all grades in the successive issues of the Roll of Members has appeared as follows:

First Catalogue, dated September 1880	Members	163	
	Associates	17	
	Juniors	9	
			189
Second " " January 1881	Members	190	
	Associates	17	
	Juniors	12	
			219
Third " " January 1882	Members	262	
	Associates	18	
	Juniors	14	
			294
Fourth " " March 1883	Members	326	
	Associates	19	
	Juniors	19	
			364
Fifth " " January 1884	Members	399	
	Associates	21	
	Juniors	20	
			440
Sixth " " January 1885	Members	514	
	Associates	23	
	Juniors	21	
			558

The loss, by death, since the last Annual Report in Volume X., has been as follows:

Daniel N. Jones	Member.
Cornelius H. Delamater	"
John Ericsson	"
Harvey F. Gaskill	"
W. H. Scranton	"
Alex. Hamilton, Jr.	"
Wm. R. Jones	"
Henry Parsons	"
John Coffin	"
Howell Green	"

The Council would also present the Report of its Tellers of Election as follows:

The undersigned were appointed a Committee of the Council, to act as Tellers (under Rule 13), to scrutinize and count the ballots cast for and against the candidates proposed for membership in the American Society of Mechanical Engineers, and seeking election before the XXth Meeting, New York, 1889.

They have met upon the designated days, in the office of the Society, and have proceeded to the discharge of their duty. They would certify for formal insertion in the records of the Society, to the persons whose names appear on the appended list, to their respective grades. There were 460 votes cast, of which 12 were thrown out because of informalities (the member voting having neglected to indorse the sealed envelope).

AS HONORARY MEMBERS.

Coode, Sir John. Eiffel, Gustave. Hirsch, Joseph.

Seventh Catalogue, dated January 1886	Members	552	
	Associates	23	
	Juniors	30	
			605
Eighth " " January 1887	Members	653	
	Associates	27	
	Juniors	34	
			714
Ninth " " January 1888	Members	726	
	Associates	35	
	Juniors	43	
			804
Tenth " " January 1889	Members	779	
	Associates	37	
	Juniors	59	
			875

AS MEMBERS.

Adriance, Benjamin.	Howell, Edward I. H.	Smith, Charles F.
Alberger, Louis R.	Hunt, Alfred E.	Taylor, Warren H.
Bates, Alex. B.	Jewett, L. C.	Tremain, E. E. G.
Blair Horatio P.	Johnson, Warren S.	Trilly, Joseph.
Britton, J. W.	Kafer, John C.	Vernon, William G.
Burpee, Geo. H.	Knous, Franklin P.	Victorin, Anthony.
Carse, David Bradley.	Meserve, John W.	Voss, William.
Christiansen, Alfred.	Naylor, Ernest W.	Wagner, John R.
Elliott, W. E.	Nicoll, Chas. H.	Wellington, A. M.
Fletcher, Andrew.	Pearson, F. S.	Wendt, Arthur. F.
Fletcher, W. H.	Pendleton, J. H.	Wheeler, Schuyler S.
Grist, B. W.	Platt, Geo. H.	Whittemore, D. J.
Hall, Willis E.	Roberts, E. P.	Wilbraham, Thomas.
Heggem, Charles O.	Roelker H. B.	Williams, J. C.
Herdman, F. E.	Sewall, M. W.	Wood, W. H.
Hibbard, Thomas.	Shellmire, W. H. Jr.	Wolf, Otto C.

PROMOTION TO FULL MEMBERSHIP.

Campbell, Andrew C. Conant, Thomas P.

AS ASSOCIATES.

Baldwin, Wm. H. Darling, Edwin. Stevens, E. A.

AS JUNIORS.

Albree, Chester B.	Jarecki, Alex. H.	Smith, Thos. G.
Basford, Geo. W.	John, Harry P.	Watt, S. P.
Dravo, Geo. P.	Lockwood, J. F.	Whaley, W. B. Smith.
Gorton, John C.	Sears, Willard T.	Worcester, Vernor F.

Respectfully submitted,

STEPHEN W. BALDWIN, }
WM. H. WILEY, } *Tellers.*

The Council would also report that it has accepted the invitation received from members resident in the city of Cincinnati, O., inviting the Society to hold its Spring Meeting, or XXIst Convention, in that city, May 13th.

At the close of the Report of the Council, the second order of business was the Report of the Finance Committee of the Society, presented as follows :

The Finance Committee of the American Society of Mechanical Engineers would respectfully report to the Council the following statement of receipts and expenditures, on behalf of the Society, which have come under their direction during the Society year, from October 15th, 1888, to November 9th, 1889. The receipts have been as follows :

RECEIPTS TO OCT. 31ST, 1889.

Initiation Fees		$1,855 00
Dues { Current	$8,310 00	
Dues { Past and Advance	755 91	9,065 91
Sales		639 35
Binding		407 65
Library Permanent		89 00
" Current		344 00
Badges		553 90
Engraving		301 19
General Printing and Stationery		28 65
Life Membership		300 00
Profit and Loss		25
		$13,584 90
Bal. Nov. 1st, 1888		391 87
		$13,976 77

There is on deposit in savings banks to the credit of the Library Fund .... $1,812 95.

EXPENDITURES.

General Printing and Stationery	$1,196 84
Reprints and Publications	3,520 80
Postage	768 41
Library	248 00
Salaries	3,163 58
Office Expenses	368 71
Engraving	843 69
Binding	440 30
Meetings	748 55
House Furniture	487 15
Badges	576 00
Traveling	134 15
Rent	997 50
Work of Committees	27 50
Contingencies	10 00
	$13,531 18
Amount deposited in savings banks to account of Library Fund	168 25
Balance in treasurer's hands, November 8th, 1889	277 34
	$13,976 77

There also remain uncollected dues from members—all of which are doubtless collectable—to the amount of $737.75.

Respectfully submitted

By the Finance Committee.

The report of the Standing Committee of the Society on the

Library was presented by Mr. C. J. H. Woodbury, on behalf of the Committee, as follows:

REPORT OF LIBRARY COMMITTEE.

The Library Committee presents herewith its Fifth Annual Report to the Society, as follows:

The plans outlined in the original report, Vol. VIth of Transactions, page 11, have been continued through the year, and with a degree of success.

Circulars were sent out in the beginning of the year with a bill for dues to those members who had not already subscribed. These circulars explained the scheme of the Committee, and were accompanied with a form of agreement requesting contributions in any of three forms:

(*a*) Subscriptions to a permanent fund in payment of $10 and upward (in installments, if preferred). To this fund there have been subscribed since the last report, from members, as follows:

M. T. Davidson	$5 00
Henry I. Snell	10 00
J. F. Firestone	10 00
Walter L. Pierce	50 00
John Fritz	10 00
E. H. Bennett	10 00
Total	$95 00

(*b*) Subscriptions to an amount of two dollars to the Fund for Current Library Expenses, payable as an increase to the Dues, and at the same time. To this there have been responses since the last report, Vol. X., from:

E. H. Bennett,	W. C. Fladd,	H. Tregelles,
Francis C. Blake,	Wm. Garrett,	C. H. Veeder,
A. W. Belcher,	H. G. Hammett,	Philip Wallis,
W. B. Cogswell,	J. C. Hobart,	W. H. Wiley,
J. J. DeKinder,	Louis R. Lemoine,	W. E. Ward,
Joe F. Firestone,	C. H. Morgan,	S. S. Webber.
W. A. Foster,	T. H. Roberts,	

There are therefore 180 members now regularly contributing to this fund by this plan of a small increase in the dues, and it is urged that others should also co-operate in the further extension of this plan, and thus induce a widespread interest in the growth of the library. The expenses against this Fund for

Current Maintenance are for binding and other incidental expenditure. Any excess, after these charges are met, will be devoted to the purchase of additional volumes. The total available annual income from this fund is now $363.

(*c*) Direct contributions of books and papers of value. Under this subdivision there have been many responses during the year.

The following list contains contributions not catalogued in the previous reports:

THE COPELAND GIFT.

Burgh's "Link Motion and Expansion Gear."
Colburn & Maw's "The Water Works of London."
"Mechanics' Magazine and Engineers' Journal."
"The Engineer," Vol. 32.
"Prac. Mechanics' Journal," Vols. 2 and 3.
Warren's "Machine Construction and Drawing."
Thomson's "Heat and Electricity."
"Prac. Mechanics' Journal," Vol. 3.
"Mechanics' Magazine" for 1851.
Wicksteed's "The Cornish Engine."
"Mechanics' Magazine," Vol. 3.
Burgh's "The Compound Engine."
"Journal of the Franklin Institute" for 1855 and 1856.
Warr's "Dynamics."
Edwards's "Catechism of the Marine Steam Engine."
Long & Buell's "The Cadet Engineer."
Hughes's "Water Works for Cities and Towns."
Bourne's "Recent Improvements in the Steam Engine."
Bourne's "A Hand-Book of the Steam Engine."
Clague's "On the Architecture of Machinery."
Peak's "Rudiments of Naval Architecture."
Bourne's "Catechism of the Steam Engine."
Larkin's "Practical Brass and Iron Founders' Guide."
Salter's "Economy in the Use of Steam."
Turnbull's "Compound Engine."
Glynn's "Cranes and Machinery."
Robinson's "Explosions of Steam Boilers."
Larkin's "Practical Brass and Iron Founders' Guide."
Colburn's "Locomotive Engine."
Buell's "Safety Valves."
C. W. Williams' "Combustion of Coal."
Bilgram's "Slide Valves."
Bacon's "Richards' Steam Indicator."
"Mechanics' Journal" for 1851 to 1857.
Johnson's "Practical Draughtsman."
Burgh's "Modern Marine Engineering."
Pugin's "Gothic Architecture," 3 volumes.
Fairbairn on "Cast and Wrought Iron."

Auchincloss's "Link and Valve Motions."
Pugin & LeKeux's "Architecture and Antiques of Normandy."
Hotchkinson's "Strength of Cast Iron."
Tredgold on "Heat."
" " "Locomotive Engines." 2 volumes. Text and plates.
" " "Steam Navigation." 2 volumes.
Ewbank's "Hydraulics and Mechanics."
Elliott's "European Light-House Systems."
Isherwood's "Engineering Precedents." 2 volumes.
"The Artisan." Several volumes.
Greenough's "Polytechnic Journal."
"The Imperial Cyclopædia of Machinery." Text and plates.
Bourne's "Treatise on the Steam Engine."
"Clark's Railway Machinery." Text and plates.
Benjamin's "The Architect and House Carpenter."
Hann & Jenner's "Steam Engine for Practical Men."
Lardner on "The Steam Engine."
Parker's "Glossary of Architecture."

Together with an extensive series of back volumes of the "Journal of the Franklin Institute" and "Van Nostrand's Magazine."

During the year the Society has been able, by the courtesy of Prof. A. MacLay, of Glasgow, to complete its series of "The Engineer," of London. Its file of "Engineering of London" is now complete, except Vols. I and II., still lacking. In the "Journal of the Franklin Institute," gaps occur as follows: all previous to 1854; from 1860 to 1869.

Of "Van Nostrand's Engineering Magazine," the series of thirty-five volumes is complete from 1869, when it was inaugurated, till the end of 1886, when its publication was merged into that of another journal.

The following is a brief *resumé* of the finances of the Library Fund:

There has been actually paid in as cash to the Library Permanent Fund and reported in previous reports of the Treasurer and Finance Committee:

For the year 1884–85		$408 40
" " " 1885–86		110 00
" " " 1886–87		145 00
" " " 1887–88		95 00
" " " " (gift of Philadelphia Committee)		206 36
" " " 1888–89		89 00
" " " " (transfer from current fund)		79 25
Interest account previously rendered	$117 08	
Interest to July 1, 1889	61 26	
		178 34
Total		$1,311 35

To the fund for current expenses the payments have been as follows:

For the year 1884–85		$164 00
" " " 1885–86		254 60
" " " 1886–87		266 25
" " " 1887–88		301 00
" " " 1888–89		336 00
Total current Expense Fund		$1,322 12
Total Permanent Fund		1,311 35
Grand total		$2,633 47
The sums which were not to be immediately expended were put in savings banks by order of the Committee, and have been there accumulating interest, as the above memorandum indicates.		
The disbursements on account of the Library Funds for the purchase of books and binding of exchanges and periodicals, has amounted in previous years to	$493 27	
Add expenditure this year	248 00	
Total expenditure	$741 27	
Transferred from Current to Permanent Fund	79 25	
		820 52
So that in the savings banks is a balance of		$1,812 95
as per the Report of the Finance Committee given elsewhere.		

The following is a list of exchanges which are continually on file in the library:

SOCIETIES, AMERICAN.

American Society of Civil Engineers. New York City.
American Institute of Mining Engineers, New York City.
American Institute of Electrical Engineers, New York City.
Associated Engineering Societies, St. Louis, Mo.
Boston Society Civil Engineers, Boston, Mass.
Society of Arts, Boston, Mass.
Canadian Society Civil Engineers, Montreal, Canada.
Civil Engineers' Association of Kansas, Wichita, Kan.
Engineers' Club of Kansas City, Kansas City, Mo.
Engineers' Society of Western Penna., Pittsburgh, Pa.
Engineers' Club of Phila., Phila., Pa.
Franklin Institute, Phila., Pa.
Indiana Society Civil Engineers and Surveyors, Remington, Ind.
Master Car Builders' Association, New York City.
U. S. Naval Institute, Annapolis, Md.

SOCIETIES, FOREIGN.

Iron and Steel Institute, London, England.
Institute Engineers and Shipbuilders of Scotland, Glasgow, Scotland.

Institution Civil Engineers of Great Britain, London, England.
Institution Mechanical Engineers of Great Britain, London, England.
Institution Civil Engineers of Ireland, Dublin, Ireland.
Ingenoirs Forenginens Forhandlinger, Stockholm, Sweden.
Liverpool Engineering Society, Liverpool, England.
Mining Institution of Scotland, Hamilton, Scotland.
N. E. Coast Inst. Eng. and Shipbuilders, Newcastle-on-Tyne, England.
North of Eng. Inst. of Mining and Mech. Eng., Newcastle-on-Tyne, Eng.
Polytechnic Society of Norway, Kristiana, Norway.
Société des Ingenieurs Civils France, Paris, France.
Annales du Conservatoire des Arts et Metiers, Paris, France.

JOURNALS, AMERICAN.

American Machinist, New York City.
American Engineer, Chicago, Ill.
American Journal of Railway Appliances, New York City.
American Miller, Chicago, Ill.
Boston Journal of Commerce, Boston, Mass.
Chicago Journal of Commerce, Chicago, Ill.
Engineering News, New York City.
Engineering and Mining Journal, New York City.
Electric Power.
Electrical Review, New York City.
Fire and Water, New York City.
Industrial World, Chicago, Ill.
Mechanics, Philadelphia, Pa.
Manufacturers' Gazette, Boston, Mass.
National Car Builder, New York City.
Power, New York City.
R.R. and Engineering Journal, New York City.
Railway News, New York City.
R.R. Gazette, New York City.
Stevens Indicator, Hoboken, N. J.
The Locomotive, Hartford, Conn.
The Locomotive Engineer, New York City.

JOURNALS, FOREIGN.

Architektu' a' Inzenyru', Prague, Bohemia.
Engineering, London, England.
Engineer, The London, England.
Electric Review, London, England.
Giornal del Genio Civile, Rome, Italy.
Glaser's Annalen, Berlin, Germany.
Indian Engineering, Calcutta, E. I.
Iron, London, England.
Industries, London and Manchester, England.
L'Industria, Milan, Italy.
Practical Engineer, Manchester, England.
Proceedings Royal Tech. Mech. Laboratory of Instr.
Stahl und Eisen, Düsseldorf.

The Transactions of the Society may also be found in the following institutions, to whose libraries they are regularly sent either as donations or in return for certain publications issued by them:

Stevens Inst. Tech., Hoboken, N. J.
Fisk University, Nashville, Tenn.
Vanderbilt University, Nashville, Tenn.
Royal Technical Institution of Research, Charlottenburg, Germany.
The Yorkshire College, Leeds, England.
Arkansas Industrial University, Fayetteville, Ark.
Bureau of Naval Intelligence, U. S. N., Washington, D. C.
Ohio State University, Columbus, Ohio.
American Institute, New York City.
Rensselaer Polytechnic Institute, Troy, N. Y.
Sibley College, Cornell University, Ithaca, N. Y.
University Library, Cornell University, Ithaca, N. Y.
University of Illinois, Champaign, Ill.
U. S. Naval Observatory, Washington, D. C.
U. S. Patent Office, Scientific Library, Washington, D. C.
U. S. Patent Office Library, London, England.
Massachusetts Inst. of Technology, Boston, Mass.
(Society of Arts.)
Conservatoire des Arts et Metiers, Paris, France.
Free Public Library, Worcester, Mass.
Purdue University, Lafayette, Ind.
University College, London.
University of Michigan, Ann Arbor, Mich.
Columbia College Library, New York City.
Lehigh University, Bethlehem, Pa.
McGill University, Montreal, Can.
Iowa Agricultural College, Ames, Iowa.

The Committee, in concluding, would call the attention of the Society to the proposed opening of its library during the evenings of the week, between the hours of 7 and 10.30, in order that the library and its contents be made as useful as possible to those whose engagements during business hours would prevent them from making use of it, conveniently, at such times. The opening in the evening is something of an experiment, involving a certain outlay for fuel, light, and service, and it will remain to be seen whether the use by the members of the facilities furnished them will warrant the continuance of the evening opening of the rooms.

The rooms will not be open on New Year's, Fourth of July, Thanksgiving, or Christmas evenings.

The President.—I would add just one word to what Mr. Woodbury has stated, to call attention to the fact that our library

fund is growing hopefully, and chiefly from the contributions of two dollars per annum from members who have notified the Secretary of their willingness to pay that small sum each year until they notify him to the contrary. In this way the fund is increasing without any trouble to any one and with a very small additional charge. All members who take interest in the library fund, and who can afford to do so, can do no better work for it than to notify the Secretary that they are willing to have this small sum of two dollars per year assessed upon them.

The Committee on Standard Flanges of Pipe reported progress.

The Committee on Duty Trials of Pumping Engines reported progress, and that they hoped their full report would be ready for the Spring meeting.

The Committee on Uniform Test Pieces and Methods of Test reported provisionally through Prof. Thurston, for the Chairman. This report embodies not only certain of the recommendations of the Committee, but also a translation made by Gus. C. Henning, reporter of the Committee, of the conclusions reached at the Conferences at Munich, in September, 1884, and at Dresden, 1886, relative to uniform methods of procedure in testing building and structural materials. This translation is the first which has been made of these resolutions and will be published at the close of the papers of this meeting as one contributed to it. The report of the Society's Committee will itself be presented later, when the Committee has been advised by certain members of the Society, who may have then had the opportunity to study the suggestions made in Germany, and so that, if possible, their recommendations may partake more of an international character.

The President, under the rules, appointed Messrs. Webster and Hill as tellers to count the ballot for officers to be elected at this time, such tellers to report later in the session.

The Committee of the Society appointed at the Erie meeting, in May, 1889, to consider the question of requesting the constitution of a govermental bureau for the recording of standards, reported, through Mr. Coleman Sellers, the following:

COMMITTEE REPORT.

To the President:

We, your committee appointed at the Erie meeting to take under consideration the subject of a Govermental Bureau for the record of Standards, beg leave to report as follows:

First. We give our earnest recommendation to the project, and advise that

it is a subject to which the Society can with propriety lend its utmost aid and encouragement ;

Second. We present herewith a resolution looking to the appointment of a committee to further the matter of Congressional legislation on the subject, and looking also to practical operations under the legislation in case the same should be secured, and we recommend the passage of the resolution ;

Third. We present herewith a resolution giving the Society's approval of the installation of such a Govermental Bureau, the object of this latter resolution being to provide the above committee with the Society's endorsement of the project, and we recommend the passage of the resolution ;

Fourth. We present herewith the draft of an Act of Congress looking to the establishment of the desired bureau, and suggest substantially such an Act as the basis of the operations of such committee.

Respectfully submitted,

James W. See,
Coleman Sellers,
Oberlin Smith.

The Committee's first series of resolutions was then presented for adoption by the Society, and was carried in the form below :

RESOLUTIONS.

Resolved, by the American Society of Mechanical Engineers :

First. That a committee of three be appointed by the President, such committee to be known as the Committee of Standards ;

Second. That when, from any reason, vacancies arise in such Committee, the President shall, on notice thereof, fill such vacancy by appointment under the Rules ;

Third. That it shall be the duty of such Committee on Standards to use all reasonable efforts to secure such Congressional legislation as will provide a Governmental Bureau of record wherein may be entered respectably recognized and approved Standards, for the promotion of uniformity in the products of arts, in technical customs, and in nomenclature ;

Fourth. That in the event of such legislation being secured and such bureau being provided for, it shall become the duty of such Committee on Standards to file, on behalf of this Society, applications for the entry of such Standards as the Society may hereafter from time to time approve for record ;

Fifth. That properly-certified bills for postage, stationery, and other expenses incurred by such Committee of Standards in its efforts to procure the Congressional legislation herein looked to, be paid out of the funds of the Society in amounts limited at the discretion of the Council.

The second series of resolutions, which was thought to commit the Society to an opinion or policy, and which would be transmitted as introducing the proposed bill to Congress, was as follows, after suggested amendments were made by the meeting and accepted by the Committee :

RESOLUTIONS.

Resolved, by the American Society of Mechanical Engineers, having more than one thousand members engaged in the manufacturing industries of the country and in allied professional pursuits:

First. The time has arrived when practical Standards of uniformity based on a common understanding are essential to industrial business and professional pursuits;

Second. A large number of practical Standards of respectable recognition and approval are now in use, but without authentic record;

Third. Systemetic procedures in the matter of recording such approved Standards would tend toward the inauguration of many other greatly needed Standards, tending toward uniformity and interchangeability of merchantable products, in improved codes and signals, and in scientific nomenclature.

Fourth. Provision should be made, under Govermental auspices, for a place of record for Standards having respectable recognition or approval regarding their fitness for such record.

The Discussion on the adoption of this series of resolutions was opened by Mr. Kent.

Mr. William Kent.—I do not wish this resolution to be hastily put and carried without a full appreciation of the fact that in passing it in its present form, we will violate an established precedent of the Society, which is, that in a meeting like this, no resolution shall be passed which commits the Society to an opinion. That question has been debated two or three times in the history of the Society, and for four or five years at least, we have had the practice and the custom established, that the Society should pass no resolution which would commit the Society to an opinion on any subject. I merely make this statement so that if members vote for the resolution they will vote with a knowledge that this resolution is an exception to the custom. I have no other objection to the resolution.

Mr. Coleman Sellers.—It seems to me that this differs from any of the questions which have been under debate heretofore. It is one which does not involve a great deal of responsibility for the members. It is one in regard to which their minds should be made up very quickly. While I perfectly agree with the wisdom of delay in legislating on matters of opinion which has been advocated and now expressed by the last speaker, I think that with justice, this subject might be acted upon at once and save some time.

The President.—I think it would be proper to qualify the statement made by Mr. Kent to this extent—that the Society has repeatedly taken ground in the past that it would not

endorse any definite standard or recommendation, and permit it to go forth to the world as the "STANDARD OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS." But no such action is contemplated in the measure now under consideration. It is simply that this Society shall certify to a Bureau of the Government that such and such a standard is worthy of record, leaving it entirely open to any one to use that standard or not.

Mr. Sellers.—This proposition, however, does not even go so far as you now state. It only announces that the Society thinks it wise to have standards, and that it would be convenient to have some place where those standards can be recorded, and the Society is committed to no particular line of action. It is not asked to endorse any opinion or method. It is more limited even than the President has shown it to be.

Prof. R. H. Thurston.—Without expressing an opinion (which I actually haven't yet formed distinctly) in regard to this whole matter, I would call the attention of the Chairman of the Committee to the first line of Section 2, in the Act which has been prepared, reading "each application for the entry of a standard must show the approval of an organized association of individuals, respectably representative of some art, industry, profession, or branch of trade, or vocation concerned with such standard, or of the head of some department of the government, etc.;" which would seem to be an indication that the Committee had presumed that societies like this would express themselves positively regarding matters of this sort.

I am not at all sure that it is not a wise thing to do. On the other hand, I have not thought the matter over sufficiently long to be convinced that it is the right thing to do; and I, for one, would prefer very much to have it go over until all members of the Society, including myself, could have time to give it more thought. I feel, myself, some serious doubt as to the advisability of proceeding with precipitation in the matter; and I am not sure that I am yet confident that our Society ought to proceed at all. I am certainly not committed against it; but I think we should move rather carefully; and I have refrained from taking some action, which I had originally proposed to take, looking to the promotion of work of the Committee, simply for the reason that I feel some doubt in regard to the whole matter, and I would like to have time to remove those doubts in my own mind.

Prof. Sellers.—I think it is always objectionable to act hastily, but of course this subject impresses me differently perhaps from the way it may impress others, having thought it over with the Committee. It is a matter in which I feel deep interest, knowing the position which America has already taken in regard to systems of standards. Every scientific society in this country occupies much of its time and effort promoting uniformity by means of standards. You know what the Franklin Institute has done, and other societies. Sir Joseph Whitworth came to this country at the time of the Exhibition in the Crystal Palace, and upon his return to England he told of the difference between America and England, and of the great facility here in obtaining work made to standard sizes at that early day. He said that there was not at that time in England a place where sashes and doors could be made to standard sizes; that every architect has his own notions of proportions, and the same was true as to the glass to be put into the sash. But feeling sure the subject will commend itself to all members of the Society, I am perfectly willing that this should remain over until Thursday, to give time for consideration.

Prof. Thurston.—If the Chairman approve, then I will make the motion that the matter be laid on the table, to be taken up as the special order on Thursday. As offering a suggestion to those members who have not perhaps thought so much of the matter as some of the rest of us have, I would remind them that this Society has taken the lead, in a certain sense, in the creation of a number of standards, but it has not resulted in the committing of the Society, or any of its members, in any matter of doubt; but it has been accomplished by appointing committees of experts, who study the matter carefully in the light of the knowledge which they themselves possess, and who report what in their opinion is a proper method of procedure, not in asking legislation perhaps, but in doing the work assigned. Such was the Committee of our Society on the testing of boilers. There we had certain members of the Society presenting simply their opinion; members who may be expected to have as much knowledge of the subject as any, and whose opinions have some weight with the profession. The report is allowed to go out, carrying the weight of the opinion of those who present it, and of those who subsequently endorse it; the Society as a body does not commit itself to any opinion, and does not take the attitude

that it has any right to express a definite opinion in regard to matters of detail of that kind.

I do not wish to be understood as objecting at all to the work of this Committee. On the contrary, I think that the object which they have in view is an eminently desirable one, and I hope to see a practical way found of attaining it. All I ask is that members may have ample time to think the matter over with care; so that each shall know what he does think, and which way he may finally best vote.

Mr. Oberlin Smith.—I second the resolution to postpone this matter until Thursday. I think it is better that we should all look over the copies of this resolution and form an opinion and be prepared to offer amendments that seem desirable; and as we have a large and representative meeting I would particularly deprecate passing it over until the Spring convention, because it is a matter of great importance that should be voted on *now.* As Dr. Sellers has remarked, it is not committing the Society in any important sense, and we have no law, at any rate, against voting on a question of this kind at our meetings. The matter in question is one that should be brought before Congress this session, early in December, so that it will not be crowded out in the Spring as so many important bills are; bills which members of Congress are perfectly willing to pass, but do not have time enough left for. If this can be settled by Thursday or Friday, and properly put before Congress early in the session, we will have a chance of its passing this winter, which will put forward this important reform a year at any rate.

[At this point the motion to lay on the table was put and carried, and the discussion was adjourned until Thursday. When the subject was resumed, the debate was opened by Prof. Coleman Sellers.]

Mr. Coleman Sellers.—It may be well to state for the information of those members who have not carefully read the proposed action of the Society that this plan of Mr. See's to organize a Bureau in Washington for the record of standards is one that does *not* commit the Society to any standard or commit them to any opinion in regard to particular standards. I do not think there can be any difference of opinion as to the one principle that is involved in it, that is, of having in Washington, at the seat of Government, some record of all the proposed standards that may be offered. We find now that for the want of some accessible published statement of the attempts of various socie-

ties and individuals to arrange and systematize their output that there is apt to be confusion, and work done by different societies and different organizations seems to clash. As for instance, at the present time I have learned since I have been in this room to-day, that while in England there is an attempt made through the Board of Trade to have a law passed making it illegal to make any attachment to gas meters other than those that are of the form, size, dimensions, pitch, etc., of threads prescribed by them for the standards of meter connections, and that the same thing was being done in America without any knowledge at all of what was being done in England; it so happens that the actions of the two peoples interested in this, almost coincide, and that there is scarcely any difference between them at all. There is, in fact, so little difference, that if at the present time the American people interested in this one question should push their system to the fore and have it established in America as the standard, there is but very little doubt that the slight discrepancies would be acceded to by the people of Great Britain, and the standard of America would be adopted in England. We might as well be in advance of the world in these things as to follow the lead of others. You know perfectly well that this Society recommended the Briggs standard for pipe threads, but still there is confusion in regard to it. It has not been thoroughly carried out. It is possible to have this all done in a better way, if any person interested in the subject knows exactly where to go for the information. Now, what is proposed by this Committee on standards is that any organized society as a society of civil engineers, or a gas society or a society of electrical engineers, or the Franklin Institute, or any other institute recommending a standard, that that standard shall be recorded in the Patent Office at Washington upon the payment of a certain fee, say a fee of fifteen dollars. It is then published in the proceedings of the patent office in the *Gazette*, so that everybody having access to the *Gazette* will know exactly what is done. It does not by law compel people to adopt these standards. Nobody is compelled to adopt them, but it is safe to trust to the good common sense of the American people to select the fittest from them all and adopt that which is the best. It enables one to know where he can find the information which is of the most importance in this particular branch. We do not ask the Government to give us such service for nothing. We say that this shall only be done upon

the payment of a fee which is more than sufficient to compensate for the record and the publication. Nor do we ask this Society to commit itself to any question except the one thing, that it is desirable to have some one place where these things can be recorded, precisely as it is when you take out a copyright. You must deposit in the library at Washington two copies of the book you publish, and everybody knows that it is possible to find in Washington a duplicate of what has been copyrighted. So if this plan is carried out, it will be possible to know at once what has been done in reference to standards by applying to the Government Department. I hope, in presenting it in that way, there will be no difference in regard to the principle involved. There may be a difference of opinion in regard to the method of accomplishing the object. It may be that the Committee has not presented the best plan. I believe some changes may be made to advantage. I have even heard that it is proposed to improve it very much by prefixing a preamble to the act of Congress, and I hope that some one will present such a preamble and do everything to make the action of this Committee stronger than it is at present.

Mr. Kent.—I am entirely in favor of the proposed Standard Bureau. I think the act to be passed by the House of Representatives and the Senate is all right, except that, if passed in its present form, the American Society of Mechanical Engineers may find difficulty some day in agreeing upon a standard, and the question might be raised whether a minority of the Mechanical Engineers should be able to prevent the adoption of a standard. Professor Sellers has said that the American Society of Mechanical Engineers has adopted the Briggs Table or system for pipe threads.

Mr. Sellers.—Recommended it, I understood.

Mr. Kent.—They have not recommended it at all. The American Society of Mechanical Engineers deliberately refused to recommend that standard, and took the same action concerning it that they did in regard to the report of the Boiler Test Committee, and refused to recommend any standard whatever of any kind or to express any opinion whatever as a Society on any subject. This question was very thoroughly discussed at the Atlantic City meeting, where the question came up of the Society approving the report of the Boiler Test Committee. It was finally resolved not to approve the report, not to disapprove it,

but to receive the report and order it printed without any expression either of approval or disapproval. The report goes before the world signed by five members who were the Committee, and the Society is in no way responsible for that report, except that it authorized it to be printed in the transactions, together with the debate on it. The same question was raised when the report of the Pipe Thread Committee was presented, and it was resolved not to approve that report, not to disapprove it, but to order the report printed. As was well stated at the time of the discussion of the Boiler Test Committee, the formal report presented by the committee of five men who are acknowledged and appointed by the Society as experts in any particular branch, will carry greater weight than an expression of an opinion by a majority vote of a meeting held in some city, at which there is not one-tenth of the Society present, and not one twentieth expert in the particular subject reported on.

The proposed Act of Congress states that an application for registration of a standard must show the approval of an organized association of individuals or the head of some department of Government or the Chancellor of the Smithsonian Institution. The American Society of Mechanical Engineers is one of these organizations. We have already established a precedent, that we will not recommend anything, will not try to force a standard or a rule or law upon any body of men whatever, or upon the country; but that the Society will appoint committees to make researches to bring in their reports, and will order these reports printed in its transactions if it thinks them worthy of it, but it will not place itself on record as trying to enforce a standard. Now, I would suggest that this act be amended by the insertion of the clause in the second line of Section 2, after the word "individuals," "or a regularly-appointed committee of such association." That will, I think, eliminate all the difficulties.

Mr. Sellers.—I would accept that amendment without any hesitation, at all; and if in our own Society we could limit the action of these proposed committees in this manner, that we should definitely authorize their publication of a proposed standard, not thereby committing the Society to it at all, we should thus assure some control over the number of standards to be adopted. That is, while we do not commit ourselves to any particular standard, we should certainly have the right to say whether one of our committees should have the right to publish

it. Now, this plan does not commit any one to the adoption of these standards, nor does it commit the Society in any way; but if this clause is added, it will add very much strength to the whole act, and at the same time free us from some embarrassment in the matter.

Mr. Geo. M. Bond.—As I understand it, the word "recognition" applies to public recognition, which would obtain before the approval of, and not directly connected with any action by the American Society of Mechanical Engineers.

Standards might be found to be of public utility before the Committee had anything to do with them, and in that way be first brought to the notice and have the approval of this Committee; so that the word "recognition," as it there appears, would not imply, by any action we may now take, an endorsement by this Society. I would also suggest that the word "such" be inserted before the word "record" in the resolution.

The President.—The action taken by this Society under our present rules would be rather in the nature of an approval of a standard as one fit for record, not necessarily as endorsing the standard *per se.*

Prof. Thurston.—As bearing upon this matter of amendment, as well as the original motion, I would suggest that the action has hitherto been: not the approval by a committee of a certain scheme, but the presentation by a committee of the best that they have been able to prepare, in the light of their knowledge at the date of their presentation; and, presenting it in that form, they simply say, "this is the best we have been able to do, and we wish to put it on record;" those who believe it desirable will accept it; those who do not, will not approve it. No individual or society is committed to the scheme; but it nevertheless represents a standard toward which all may work as may be deemed best. The result will naturally be that, if we do succeed in finding a method that is practical and useful, those most interested in the matter will accept it gladly, and it would become by general practice an accepted standard; and this is the only way by which a standard can be made. It requires no endorsement by us, and should have none, I think.

Mr. Philip R. Voorhees.—The motion before the Society, if I apprehend it correctly, is to adopt the resolution proposed by the Committee. It seems that there is a diversity of opinion as to the propriety of passing this resolution as a resolution, in

view of the fact that it commits the Society to an opinion. It has occurred to me that perhaps a solution of the difficulty might be reached at the proper time, in discussing the act itself—that then the substance of this resolution, which certainly very clearly puts the necessities of the case, might be embodied in a preamble to the act itself. If this is done, it might avoid debate as to the propriety of committing the Society to the expression of an opinion. At the proper time I would therefore move that the substance of the resolution which I have embodied in almost its identical language, be inserted as a preamble to the act. Then the source from which such language emanates will be known, and we will have the advantage of setting forth the necessity so clearly put by the gentlemen of the Committee.

I have some hesitancy in making the motion in this form, because I must move this in lieu of the resolution now before the meeting, nor did I want to take so advanced a position until there had been some further discussion on the resolution, but I will make the motion with this explanation to the gentlemen assembled.

I will move, therefore, that in lieu of this resolution, that its substance be embodied in the act as a preamble thereto, which reads thus:

"Whereas practical standards of uniformity based upon a common understanding are essential to industrial business and professional pursuits," etc.

The President.—If the meeting will accept a suggestion from the Chair, the simplest procedure will be, first, to pass the resolution which has been offered by Dr. Sellers, then after that to pass a motion to the effect that the resolution be incorporated as a preamble in the draft for the act of Congress to be submitted by the Committee for Congressional action. It would then read in the preamble, "Whereas, the American Society of Mechanical Engineers has adopted a resolution to the effect that in the opinion of this Society, having over one thousand members," and so on, following with each of the clauses of the resolutions as passed.

After one or two verbal amendments were suggested by the members and accepted by the Committee, the vote was taken on the amended resolutions as they appear, and the resolutions were passed. The act, as amended by prefixing the preamble

proposed by Mr. Voorhees and accepted by the Committee, is in their hands for presentation to Congress. Its substance will appear in their later report.

The Chair announced as the Society's Committee on Standards under the resolutions:

Messrs. James W. See.................................... Hamilton, O.
Coleman Sellers.................................Philadelphia, Pa.
Philip R. Voorhees.................................New York City.

The amendment to the rules of the Society proposed by Mr. Woodbury and duly recorded at the Erie meeting in May, 1889, under the rules for amendments, and which had been distributed in printed form in advance of the meeting, was now taken up and passed. The amendment, as modified from its first form, to conform to the requirements of the laws under which the Society is incorporated, is as follows:

PROPOSED AMENDMENT

TO THE

RULES OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

ARTICLE 20.

The affairs of the Society shall be managed by a Council, consisting of a President, six Vice-Presidents, nine Managers, and a Treasurer, who shall also be the Trustees of the Society.

All past (Ex) Presidents of the Society, while they retain their membership therein, shall be known as honorary Councillors, and shall be entitled to receive notices of all meetings of the Council, and may take part in any of its deliberations; they shall be entitled to vote upon all questions except such as affect the legal rights or obligations of the Society or its members.

ARTICLE 21.

The members of the Council shall be elected from among the members and associates of the Society at the annual meetings, and shall hold office as follows:

The President and the Treasurer for one year; and no person shall be eligible for immediate re-election as President who shall have held that office for two consecutive years; the Vice-Presidents for two years, and the Managers for three years; and no Vice-President or Manager shall be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

The President.—I think it proper to say a word further in explanation of the change which has been made, the object of which is to give continuity and permanence to the Council to a greater degree than exists at the present time. The past Presidents of the Society comprise those members who have had the greatest experience in its administration at one time or

another. It is believed that the addition of the past Presidents to the Council will contribute both qualities I have mentioned, namely, continuity and unity of policy and precedent as well as knowledge of and experience in its past doings.

Ex-President Coleman Sellers now took the Chair, by request of President Towne, in order that the latter might present to the meeting the following preamble and resolutions:

Whereas, There are now in existence four associations of American engineers, each representing one branch of the profession, and each serving a useful purpose; and

Whereas, Many members of the existing societies are desirous that a new organization should be created which, without superseding or disturbing those already in existence, will be broader and more comprehensive than any of them, and thus more representative and national; and

Whereas, It is believed that the result thus outlined may be accomplished without any encroachment on the autonomy of the existing organizations, and without any impairment of their distinctive character; and

Whereas, It is especially desirable, in view of the contemplated international exhibition in 1892, that there should be some organization broadly representative of American engineers, and thus qualified, in their behalf, to receive and entertain the foreign engineers, who, it is hoped, will then visit the United States; and

Whereas, The desirability and feasibility of closer relations between the members of the existing organizations were demonstrated by the experience of those who took part in the joint excursion of American engineers to Europe in the summer of 1889.

Therefore be it Resolved:

1. That the AMERICAN SOCIETY OF MECHANICAL ENGINEERS views with favor all efforts tending to establish more intimate relations between the existing engineering societies, and to develop a national organization of American Engineers;

2. That to promote this object the Council be hereby requested to communicate the substance of these resolutions to the officers of the other engineering societies, and to invite them to unite in a discussion of the subject, and in an endeavor to reduce it to proper form for submission to the consideration of the membership of each society;

3. That to this end the Council be further requested to appoint a committee of three members to represent this Society in any such conferences, to consider the whole subject herein referred to, and to report the result of their labors to the next meeing of the Society.

Mr. Towne.—Several times during the past three or four months, both in this society and in our sister societies of civil and mining engineers, mention has been made of the possibility of some closer union between the engineering societies of the United States. The suggestion arose most naturally from the necessity for such action among those members of the several

societies who took part in the excursion to Europe last summer. The ease with which a union was accomplished then, and the happy results which followed from it during the trip, suggested to many of us that something of the same kind might be done upon a larger and more permanent scale. With that in mind, the foregoing preamble and resolutions were offered: I hope there will be some discussion of this subject. It is one of large scope. I think the meeting would be very glad to hear the opinions of some of those older members of the Society who took part in the excursion of last summer. Professor Thurston, for example.

Prof. R. H. Thurston.—Unfortunately, for me, I was not with the committee of three hundred and sixty during all its peregrinations of the last summer. But I know something about what was done by them. The suggestion which has just been made, in these resolutions, is one which was first made in substance many years ago. It was one of the many great plans for the benefit of the profession which were suggested and urged by Alexander Holley. Long before his death, he was endeavoring to secure the approval of the active and leading members of the several societies of engineers for the carrying out of just such an idea, in the general form of that which has now been presented. His belief was that if he could secure a union of the active and the leading members at least, of all the great societies; such as would constitute an academy of engineering, or an institution of engineering; some association, in definite form, of all the best men of all the societies, and as many others as could be induced to join,—that the result would be the securing of a bond of sympathy and of union in professional work among these societies which could not fail to have very valuable results. If I am not mistaken, he secured the appointment of a committee, in each of two or more of the societies, whose duty it was to endeavor to secure action looking toward such union. The plan never took complete form; but it was talked over by Mr. Holley, and by his friends, frequently for a period of several years before his death, and has as frequently come up since, in various connections, in the several societies. The experience of that joint convention which was, by an accident as it were, held last summer in the visit of the several societies of engineers going together to their friends on the other side of the water, led to the revival of this idea in substantially the same form, and that, as we have just been told,

has led to its presentation here. It seems to me that any one who has given any thought to the great work which lies before us, as an association of specialists in engineering, will be prepared to approve the general scheme. What form it may be practicable to give it, cannot to-day be decided. But a properly constituted committee would, I have no doubt, be able to present some scheme, in time, when the matter had been sufficiently thought over by the members of the various societies, which would lead to some such action as is here proposed, and which seems to me to outline one of the grandest purposes. Such a union would be of enormous advantage to the members of the several societies, and to the societies as independent bodies, as well as to the profession at large, and to the whole country. I should myself be very strongly inclined to urge that such a committee be formed, and I shall hope that if a committee be so constituted here, it will find committees from other societies ready to meet it more than half way in its work, and that ultimately we may find a way of bringing all of these great societies into common union for all those purposes which are appropriate to all alike.

Mr. Oberlin Smith.—As one of the committee spoken of, who went abroad last summer, I want to testify to the feeling which seemed to be general among the members of that party, in favor of just such a combination or union of forces, as this. I know that the social and professional intercourse between members of the different societies, during their trip, has greatly forwarded any movement of this kind which may come up. Whether we shall be able to do it immediately or not, there certainly is more chance than there was a year ago. I am heartily in favor of something of the kind, though I did not know it was coming up this morning. I am very glad that it has come up. I think those of us who have been to England, France and Germany, and have seen the condition of the engineering societies there, will be still more in favor of it, on account of the example they are setting us—especially in the case of the British Institution of Civil Engineers, having nearly five thousand members. In this society, although the name is "Civil," the members are not all, by any means, nor a majority of them, what we know as civil engineers here. They are simply non-military engineers. They are engineers of all branches—men who build bridges, roads and canals; and men who make steel, and who

build boats and engines; men who do engineering work of all kinds, who are mining engineers, hydraulic engineers, and electrical engineers. These men together form the "Institution of Civil Engineers," all working together in a civil way, trying to aid each other in every way they possibly can, and forming a great society of power and influence—one, moreover, which is not split up into warring factions, in any way whatever. I do not see how it would have been possible for American engineers to receive the magnificent ovation which they did receive, had the invitations come from small societies merely. I do not see how we, in 1892 (or 1893, or 1900), can expect to invite these men over here properly, and have a "respectable recognition and approval" of that invitation by them, unless it comes from some common source, some kind of a union of societies. Of course, there might be a temporary committee appointed from the different societies, to invite them; but it would not have the force which a more regularly organized Commission, or Joint Society, or Academy would have. It is, I think, particularly desirable at this time that we should do something of the kind, and that no time should be lost, so that a definite policy may be settled upon at the next spring and summer meetings of the various societies, in order that such an invitation may be sent in time, and plenty of time. Of course, this is not the only object in forming this union. We must look to our own interests, the interest of American engineers in every way, and whatever we do now, rather in a hurry perhaps, that we may extend this invitation abroad, will redound to our own benefit during all the coming years.

Mr. O. C. Woolson.—I understood Mr. Towne to suggest that the committee consist of three from this Society. It strikes me that that is entirely too small. Large bodies move slowly, but doesn't it seem that three is rather a small number to consider a matter of this great importance?

Mr. Towne.—To answer the suggestion: Mr. Woolson has said very correctly that large bodies are awkward, and are not rapid in their work. If each of the other societies responds, we will then have a committee of twelve, which I think is quite as large as could do any effective work, and it is to be borne in mind that the work of the committee is by no means final or conclusive. It is simply preliminary. Its object will be merely to ascertain what are the views ot the four societies, and then

for each to report the result of its investigation to its own society. It is provisional work, rather than anything final.

Mr. J. F. Holloway.—As one of a great many who were obliged to stay at home last summer, I did not have an opportunity of knowing, as those who were abroad did, of the necessity for such an organization. I think, very likely, they found it was a very great improvement over there, to have all the engineers grouped together. The definition which Mr. Smith gives of the term civil engineers is somewhat new to most of us, because that term in this country denotes a distinct class of engineers, and I think the name, British "Institution of Civil Engineers," conveys by its title the idea that it is composed of the same class of people as are the civil engineers in this country; but if they are an association of all engineers, I can understand that they must have a very large power, and exert a very great influence. In this country the distinctions are quite different, and we are classified by the special profession which we follow. And I, for one, while I heartily agree with the idea of a committee to investigate the matter, and to formulate some plan, and present it to the Society, have, I confess, very great respect for the growing society—the American Society of Mechanical Engineers. They have done a great deal in the short time they have been in existence, and we have great hopes of their future, and I do not want to see them sunk into another society in which they shall not have in the future the position that they certainly would be entitled to, and which they already have in the industrial progress of this country. I agree, as I say, in the idea that we should investigate the matter, and if any possible plan can be thought of, in which we can all in a general way join together in some one society, well and good. But I still think we have our own place. We have our own business to do; and I think we are perhaps as well fitted as anybody else to fill our own place and situation, and meet the opportunities that are constantly presented to us. The place of the Mechanical Engineers is a very broad one, and a very wide one, and I think that this society is coming up rapidly to take that place, and I do not want to see it overshadowed by any other. (Applause.)

Mr. Oberlin Smith.—I speak again only because Mr. Holloway misunderstands me. I had no idea of suppressing this society or any other, nor of sinking them into a new and common society. I simply spoke of the organization over there which was enabled

to invite us as they did and treat us as they did. I do not see that any academy of engineering or whatever it may be, as proposed by Mr. Towne, and elsewhere by Mr. Kent, will take away the individuality of the other societies at all. It will only help them in their work. Of course, we do not yet know in what form this matter will be finally crystallized.

Another point: in speaking of the word "civil" as meaning non-military, I only referred to the origin of the word, and did not mean to convey the idea that we do not want military and naval gentlemen with us. Indeed, some of our most accomplished and practical brethren are military and naval engineers; and it happens also that they are largely engaged in helping us in the arts of peace. Our society should embrace all, whether they are known as civil, mining, mechanical, electrical, naval, or military engineers. What we want in our coming greatest and most important society of all, are engineers—simply engineers. We do not want all the adjectives, some of them not very logical in their derivation.

Mr. Jesse M. Smith.—I am heartily in sympathy with this union proposed by the resolution of Mr. Towne, and I would like to see it extended even farther than Mr. Towne has suggested. Let it include all scientific societies, that the day may arrive when we shall have in New York or some other metropolis of the country a home for scientific societies where every society can come in its turn and hold its conventions, and have a home which is respectable, and a library which shall be worthy of consultation and everything really desirable. The idea which Mr. Oberlin Smith has stated, as regards the Institution of Civil Engineers of Great Britain, has béen carried out to a still further extent in the Society of Civil Engineers of France. The Society of Civil Engineers of France is made up not only of what are the civil engineers,—that is, engineers from civil life,—but the engineers from civil life are very largely engaged in military and naval pursuits, as is instanced by M. Garnier, who has done so much for the present armament of the French cruisers and the land guns, as was shown by the Paris Exposition. I feel, therefore, that there should be a society of engineers in this country —not of mechanical engineers, civil engineers, mining engineers, or any other, but of engineers—not that this Society should overshadow all the others, but that each specialty shall have its own work to do and do it as it has done in the past. And that there

shall be a society over all the others consisting of all the others, and that they should be united so that their actions shall have weight, so that when we are in a position to invite our brothers from the other side that we will do it as a national society of engineers, and that the full weight of brotherhood can be extended to them in that way.

Prof. Coleman Sellers.—When I was proposed for membership in the British Institution of Civil Engineers, I made the objection that I was not in the ordinary sense of the term a civil engineer, having been most of my life engaged in mechanical engineering. Sir Frederick Bramwell, who was vice-president of the society, pointed out to me that by the constitution of the society there was no such limit as that usually applied to the term "civil engineer," and he showed me, too, that very many of their presidents in succession had been the great mechanical engineers of England. The same men had also been presidents of the Institution of Mechanical Engineers of England. More than three-fourths of the papers read before the so-called Institution of Civil Engineers of England are papers devoted entirely to questions of dynamical engineering, and the succession of lectures delivered before them has been such as has not been of use to the bridge builder and constructor of railroads so much as to those who had to do with the matter in motion, and in view of the fact that the so-called Institution of Civil Engineers in England is an association simply of engineers, it is rather unfortunate that any adjective is attached to the title at all.

Mr. Towne.—We have with us Mr. R. W. Pope, the secretary of the American Institute of Electrical Engineers, and we would all be glad to hear briefly from him the sentiment on their part in regard to the matter.

Mr. R. W. Pope.—I have been secretary of the Institute of Electrical Engineers since 18 5, and, while of course I have no authority to speak as representing that body in an official sense, I am well aware of the sentiment of a great many of our officers and members in regard to the subject before you. There are some points in relation to this matter which have not been spoken of, and are not recognized as they should be. One of the most important is the fact that engineers and scientific men as a body have not that influence in public matters that they should have. I refer more particularly to municipal affairs. It appears to me, and has for some time, that municipal government to a

great extent should be a matter of engineering rather than of politics. In national affairs, however, it is proper to consider political and economic questions. In the local government of cities, the condition of the streets and water works, the public lighting, underground work, and all things of that nature are engineering problems, and we are well aware that when any of these matters come up before the ordinary Common Council or Board of Aldermen, composed of politicians, they do not grapple with these questions as an engineer would. Speaking of engineers particularly, I would say that, although there are various branches of them, each branch borders on the other. There are civil engineers and mechanical engineers and electrical engineers and mining engineers, and there are architects, whose work approximates that of the civil engineer, and who should have some knowledge of electricity, sufficient at least to enable them to specify the proper material to be used, instead of simply insisting on the use of lead-covered wire, as is now frequently the case.

You have in your society about twenty-five of our members—twenty-five electrical engineers, who are also mechanical engineers. That is not such a very large number, but it is well understood in electrical engineering that there can be no thoroughly first-class electrical engineer, unless he is well grounded in mechanical engineering. He must understand mechanical engineering in order to become thoroughly equipped to practise his profession, and it also behooves mechanical engineers to give attention to electrical matters in connection with their own work. The movement in question is one which appears to me to be in accordance with the tendency of the times, which is to concentrate work of various descriptions. A complete engineering library in New York should cover all electrical subjects and all mechanical subjects, and what we consider civil engineering subjects. In order to make an electrical library, we should have a library of mechanical engineering. But it is a matter of twenty-five or thirty thousand dollars to get even a good electrical library. I have the same files of technical journals in my office that your secretary has in his, and nearly the same that civil engineers have. I went so far in this matter, at one time, as to suggest to your secretary the propriety of organizing in New York what I termed a society of societies—a working board of officers of the different engineering societies—in order to

exchange views as to the conduct of society work, and the introduction of uniform methods, so that in case of any business coming up, similar to the European excursion last summer, the different boards could work together in harmony; the machinery being already in existence, it would be merely necessary to set it in operation. Now, I think we would be wise in profiting by the experience of our British brethren. As I understand it, the Institution of Civil Engineers is the leading body, other organizations being the mechanical engineers, the electrical engineers, etc. The Institution of Electrical Engineers in England numbers over one thousand members. It has a greater membership than your society of mechanical engineers here. They meet in the building of the Civil Engineers, and are in harmony with them. Some of the best electrical papers read in England have been read before the Institution of Civil Engineers. You will see, therefore, that a leading society, even as strong as the British Institution of Civil Engineers, does not overwhelm the others. The others are strong, in spite of there being a leading society. I think, gentlemen, that this is a question requiring thorough examination, and I feel at liberty to say that whatever may be the result, the American Institute of Electrical Engineers would be very glad to send a committee to consider it.

Mr. W. S. Rogers.—I have thought, for a long time, that the time was not far away when the electrical engineer and the mechanical engineer would be drawn very closely together, and I am impressed with that more and more every day. If there is anything like a plan proposed to organize a new society, I can mention some experiences with profit. I received a letter a short time ago from a young man, and incidentally he told me that he did belong to one of the societies, and that he did not like it. He said it seemed to him that what they were after was his fee, and what he was after was knowledge. He was not getting it as he wanted it. He went on to state that he was trying to fit himself to become a member of the American Society of Mechanical Engineers, as he thought there he would gain the counsel and assistance and knowledge which would fit him to do some good in this world. I can only hope that in planning to form this new society any committee will go very slowly and go carefully, and let the American Society of Mechanical Engineers hold the leading place or equal leading place in it. At the same time, I don't wish to be understood as being opposed to any

organized method of uniting the various societies, provided it helps in practical knowledge, and the American Society of Mechanical Engineers is properly recognized.

Mr. Wm. Kent.—There is one phase of this subject which has not been touched on. I believe that the American Society of Civil Engineers are trying to enlarge themselves by annexing the various local societies in Cleveland, Pittsburg, St. Louis, and other places. I see in the papers that those societies are considering the desirability of affiliating themselves to the American Society of Civil Engineers as chapters. These local societies are composed of mechanical, civil, electrical, and all the other kinds of engineers in one body. I understand that the American Society of Civil Engineers propose to consolidate these various societies into the civil engineers as their chapters, and so become a grand society like the British Institution of Civil Engineers; but I will not raise the question whether that will be as desirable as the proposed form of the new society, which has been brought out in the resolution by Mr. Towne.

I had the honor, in 1886, to read a paper at the meeting in Buffalo of the American Association for the Advancement of Science, entitled: "A Proposal for an American Academy of Engineering," and in which this whole subject was treated just about as we have heard here. It was published in *Van Nostrand's Magazine* for October, 1886. It discusses this question of an academy of engineering composed of the best members of the three societies—the mining, mechanical, and civil engineers' societies—and all others—the army engineers and naval engineers, the sanitary and electrical engineers. One of its objects was to promote harmony between the naval and army engineers and the engineers in civil life.

Mr. Towne.—Use has been made by one or another of the speakers of the terms "merging" or "consolidation" or "union" of the societies, implying that the existing organizations would, in some way, be obliterated. If the resolution is carefully read, it will be seen that nothing of that kind is contemplated. On the contrary, the statement is made that the autonomy of the existing societies shall be most carefully preserved, and the conviction expressed that it is possible, while preserving that autonomy, still to have something which will effect a union of members of the various societies, and something which will be more thoroughly national than anything we have at the present time.

But the scope of the whole motion is simply that a committee be appointed, that the other societies be invited to confer with that committee, and that the committee report at a later meeting to the society whether it deems any action desirable, and if so, of what kind, so that the matter may then come before the society in form for clear understanding and discussion.

The resolution was carried. The Committee was subsequently appointed to consist of Messrs. Towne, Thurston, and Sellers.

At the close of the business session of the first morning, the professional papers were taken up and were read in abstract, where full presentation would have required more time than the five-minute limit allowed in cases where the papers had been printed and distributed in advance.

The paper by Prof. V. Dwelshauvers-Dery, of Liege, Belgium, honorary member of the Society, on the "Properties of the Vapor of Water," was presented by Prof. R. H. Thurston and discussed by Messrs. Spangler and Denton; that by Prof. Horace B. Gale, on "Theory and Design of Chimneys," was discussed by Messrs. Babcock, Denton, Kent, and Peabody; that by Prof. R. H. Thurston, on the "Philosophy of Multiple Cylinder or Compound Engines," was discussed by Messrs. Denton, Ball, and Wheeler.

After these papers, the Secretary made some announcement as to the excursions of the week, presented the invitation from the Electric Club of New York to have the members use the rooms of that club during their stay, and also one from Mr. C. J. Field, of the Society, to have a party visit the plant of the Edison Company, in Brooklyn. A recess was taken till the afternoon.

The second session for papers was convened at half-past two o'clock in the same place. Prof. C. H. Peabody read his two papers together, on "Flow of Steam Through Orifices," and "An Experimental Study of Different Types of Calorimeters," and they were discussed by Messrs. Babcock, Faber du Fauer, Spangler, Denton, and McBride; that by Mr. L. S. Randolph, on "Cost of Lubricating Car Journals," was discussed by Messrs. Denton, Woodbury, Almond, Partridge, Rogers, and Dick; that by Mr. E. F. C. Davis, on "Steam Pipes for Collieries," by Messrs. Nason, Allison, Sweet, Smith, Minot, Faber du Fauer, Woolson, and McBride; that by D. K. Nicholson on "Rolling Steel Rails," by Messrs. Hunt and Gannt; that by Mr. C. J. H.

Woodbury, on "Methods of Reducing the Fire Loss," by Messrs. Grinnell, Rogers, McBride, Sweet, Towne, Laforge, Campbell, and Gobeille. After announcements, the meeting adjourned till Thursday morning.

THIRD DAY, THURSDAY, NOVEMBER 21ST.

The third session for papers was called to order at No. 12 West Thirty-first Street, at ten o'clock in the morning.

The report of the tellers to count the ballot for officers for the ensuing year, was as follows:

			Votes.	
For President	Oberlin Smith	received	426	—scattering 7
" Vice-Presidents	Joel Sharp	"	432	
" "	Geo. W. Weeks	"	436	
" "	De Volson Wood	"	443	— " 2
For Managers	Jas. E. Denton	"	440	
" "	C. W. Nason	"	432	
" "	H. H. Westinghouse	received	436	—scattering 1
For Treasurer	Wm. H. Wiley	received	438	
	Informal votes rejected		13	

J. H. WEBSTER, }
WM. HILL, } *Tellers.*

The papers of the session were: by Mr. F. W. Parsons, "Indicator Rigging for Compound Engines," discussed by Messrs. Spangler, Sellers, and Smith; by Mr. C. T. Main, on "Comparative Cost of Steam and Water Power," discussed by Mr. S. Webber; by Prof. D. S. Jacobus, on "General Solution of the Transmission of Force in a Steam Engine," discussed by Messrs. Denton, Oberlin Smith, Bond, and Scott A. Smith; by Mr. S. J. MacFarren, on "Street Railway Car Gear for Modern Speeds," discussed by Mr. Rogers.

After these papers and business, the recess was ordered till the afternoon.

The fourth and concluding session for papers was called to order at half-past two in the same place.

The papers were those by Professor Webb, on "The Comparison of Indicators," discussed by Messrs. Spangler, Suplee, and Thurston; by Mr. Holloway, on "How to Use Steam Expansively in Direct-acting Pumps," discussed by Messrs. Suplee, Webb, Denton, Thurston, Emery, and Hague; by Prof. W. H. Bristol, on "A New Recording Pressure Gauge," illustrated by

apparatus, and discussed by Messrs. Hague, Edson, and Wolff; by Mr. Oberlin Smith, on "Graphical Analysis of Reciprocating Motions," discussed by Messrs. Emery, Bond, Jesse M. Smith, Leavitt, Spangler, and Suplee; and the two papers of Prof. J. E. Denton. The one on "Indicator Cards of the Pawtucket Engine," was discussed by Messrs. Emery, Hague, Oberlin Smith, Wolff, Henthorn, Webb, Thurston, Wood, and Kafer; and the other, on the "Performance of a Double Screw Ferry-Boat," was illustrated by lantern-slides, and was preceded by an introduction by Col. E. A. Stevens, as associate-author. This paper was to have been read on the boat whose performance it recorded, but on account of an accident to the boat it was put at the close of this day's session. It was discussed by Messrs. Wood, Taylor, Emery, Kent, Hague, Holloway, Webb, and Hyde. It had become so late before the beginning of this paper, not originally allotted to this session, that before its presentation the president called for certain matters of business of whose presentation he had been advised. The following series of resolutions were then presented, seconded, and carried with acclaim:

To the Local Committee:

Realizing that the success of this meeting is largely due to the efficient services of the local committee, whose untiring efforts have caused so much pleasure to those members of the Society who have been so fortunate as to be in attendance; therefore be it

Resolved: That we tender our heartfelt thanks and appreciation to the members of the local committee, and especially to Mr. James F. Lewis, whose skill as a navigator brought us safely through Hell-Gate, after having given us a glimpse of the lurid fires beyond,—at Whitestone.

To the Patronesses of the "House-warming":

The visiting members and guests of the American Society of Mechanical Engineers feel that the brilliant success of the "house-warming" reception, at the new rooms of the Society, was due to their presence and refining influence.

We desire to express our appreciation of the results of their taste in the artistic arrangements of the rooms, and trust that future meetings will be graced by the presence of the ladies in yet increasing numbers.

To the Engineers' Club:

The American Society of Mechanical Engineers hereby expresses to the Engineers' Club its appreciation of the courtesies on the occasion of its twentieth meeting; to the board of managers, for the reception at the club-house; and to the house committee, for the extension of the privileges of the club.

We avail ourselves of this opportunity to extend fraternal greetings to the Engineers' Club, with best wishes for its continued prosperity.

To Henry Morton, Ph.D.:

It is always pleasant to acknowledge courtesies extended by others, and

doubly so when the host is so honored and distinguished a member of our organization as is the president of the Stevens Institute of Technology.

It gives us the greatest pleasure to acknowledge, at this time, the hospitality of Dr. Morton, in entertaining the members of the Society and their friends on the excursion in New York Bay, and this will be added to the remembrance of previous hospitality received at his hands.

To the Board of Managers of the American Institute Fair:

The American Society of Mechanical Engineers wish to thank you sincerely for the generous invitation extended to its members to visit the fair of the Institute, and witness one of those exhibitions, which have so long fostered the mechanic arts.

To Col. E. A. Stevens, President, and C. W. Woolsey, Superintendent of the Hoboken Ferry Company.

For their courteous invitation to make an excursion on the new double-screw ferry-boat *Bergen*, the American Society of Mechanical Engineers desire to express their thanks.

To Lieut.-Col. Wm. R. King, U.S.A., and Officers of the U. S. School of Engineers and Experimental Station at Willett's Point:

The members of the American Society of Mechanical Engineers desire to make acknowledgment of your attentions in extending to our society an opportunity of examining the Museum and the apparatus for the application of science to the art of defense.

As members of a society of civilians we wish to express our profound gratitude for those arrangements which caused the explosion of dynamite to be made at safe distances from our honorable bodies.

To W. F. Pinkham, Esq., and the Central Forge Works:

The American Society of Mechanical Engineers extends its thanks for the opportunities given to its members of witnessing the operations carried on at your establishment.

To the President and Directors of the Singer Manufacturing Company:

Among the various incidents connected with the meetings of our Society, the visits to manufacturing establishments are always instructive and entertaining, and we cannot too fully express our appreciation of the courtesy which makes such visits possible.

We desire to extend to you our fullest thanks for the privileges afforded us to examine your great establishment, and to convey our acknowledgment of the value of the opportunity thus afforded us.

To the Electric Club:

The American Society of Mechanical Engineers desire to express their sincere thanks to the Board of Managers of the Electric Club for their hospitality in extending the courtesies of the Electric Club, during the twentieth meeting of this Society.

It is a matter of congratulation that those who have scientific, mechanical, and commercial interests in applications of electricity, are provided with such an eminently useful organization as the Electric Club.

To Dr. Herbert G. Torrey:

The American Society of Mechanical Engineers desires to acknowledge with thanks, the invitation of Dr. Herbert G. Torrey, Director, to visit the United States Assay Office, during the twentieth meeting of the Society.

To C. A. Griscom Esq., President of the Inman and International Steamship Company:

The American Society of Mechanical Engineers desire to give most sincere thanks for the invitation to visit the steamship *City of Paris*, and for the hospitality gracing the occasion, which also affords to many of our members an opportunity to revisit this queen of the Atlantic fleet, of which they have such enjoyable recollections.

At the close of these resolutions, the following was presented and adopted:

Resolved, That this Society approve of the holding in this country, in 1892, of a World's Fair commemorative of the four hundredth anniversary of the discovery of the country by Columbus.

After this resolution had been put, the Chair spoke as follows:

The President.—With this session terminates the twentieth meeting of this Society, and it marks the close, also, of the most successful year of its existence. One or two words I would like to add as to our future development, and especially in regard to the preparation of papers for our meetings. It is urged again, on behalf of the Council, that a larger proportion of our members should participate in the preparation of papers to be read at our meetings, and especially that these papers should embody the results of personal practice or experiment. We wish to strengthen that side of our transactions which relates to the accumulation of actual experience in practice, or the results of actual experimental investigations, to supplement the other side, which relates to the theory of our profession. It is also well, I think, to call attention to the great value in such papers of having, at their close, a brief summary giving their results in a condensed form. One or two of the papers presented at this meeting have had an appendix of this kind, and I think you will agree with me that such a summary adds greatly to the value of any scientific paper, especially for future reference.

It should not be forgotten that three years ago the Society decided that its sessions should be open for the reception and discussion of papers relating to economic subjects, as well as those relating directly to engineering, and that such papers of

that kind as have been presented heretofore have proved interesting and valuable. It is to be hoped that others of the same character will be presented hereafter.

It is desirable, also, that there should be a more general participation in the discussion of papers, and that a larger number of the members present at each meeting should take part in the oral discussions of the papers submitted by members. In that connection I will refer to the device which has caused some amusement at this meeting, the alarm bell of the Secretary, and will recommend that it should be continued at our future meetings simply as an automatic reminder of the lapse of time. All of us, when we get on our feet, are apt to lose consciousness of the passing of time, and in that way it becomes very difficult for the presiding officer to hold the members to the rules adopted, the wisdom of which rules has now been proved by three years of use at six successive meetings. It has been demonstrated beyond question that a close adherence to those rules means not only fair play, by giving the authors of papers coming late on the docket, and those who wish to discuss such papers, opportunity to do so, but also tends to make the whole of our sessions more interesting and valuable.

Finally, I wish to thank the members of the Society for the cordial support they have given me during the past year, to assure them of my warm appreciation of it, and to express the hope that my friend and successor, Mr. Smith, may have a continuance of the same kindness at their hands. I will now announce that, after the reading and presentation of views relating to the following paper, the twentieth session of the Society will close.

The suggestion has been made that a committee of three be appointed to call on his Honor, Mayor Grant, to inform him of the action that has been taken here.

Mr. Oberlin Smith.—I will make a motion that a committee of three be appointed to wait on the Mayor of New York, and present to him the resolution just passed regarding the international exhibition.

The motion was seconded.

Mr. Hague.—It occurs to me that we have several Western members, and it might be well to present the resolution to the Mayor of Chicago as well.

The President.—I presume it is the wish of those who have suggested the matter that the interest of our society in this

project should be communicated to all who are endeavoring to bring about the holding of an international exhibition.

Mr. Oberlin Smith.—I accept the amendment for St. Louis, Chicago, Washington, and the centre of population.

The President.—The question is now upon the appointment of a committee at this end of the line.

The motion was carried, and the Society adjourned at the close of the final paper.

Excursion Days.

The week of the Convention was mostly rainy, but on the two days devoted to excursions the sun shone brilliantly.

Wednesday, November 20th, was devoted to a trip on the East River, of New York, to Willett's Point, on Long Island. There the Society were the guests of Lieut. Col. W. R. King, U. S. A., who is at the head of the post and in charge of the governmental training and experiments on harbor defense. The party was escorted over the laboratory and museum buildings and to the gun which is wound with electric cable to make it into an immense electro-magnet. Luncheon was served on the boat, and after it was over, three torpedoes were fired under water near the wharf, that the enormous effect of such explosions might be witnessed. On the return, a stop was made at the works of the Central Forge Works, at Whitestone, where the members were guests of Mr. C. F. Pinkham. On Wednesday evening, by the invitation of the Board of Managers of the Engineers' Club of New York, a reception was tendered to the Society at their club-house, at No. 10 West 29th Street. During the evening a handsome collation was served.

On Friday it had been planned that the screw ferry-boat *Bergen* of the Hoboken Ferry Co. should convey the party for a trip on the Hudson River, which would also give the members a chance to inspect the working of the boat. The failure of one of the boiler-furnaces, however, made this entertainment impossible, and instead of it, the Society accepted the invitation of the Singer Manufacturing Co. to visit their sewing machine manufactory, and the plant of the Babcock & Wilcox Co., building boilers, at Elizabethport, N. J. The same steamer (*Laura M. Starin*) conveyed the party to the Company's wharf, and after luncheon on the boat tendered to the party by Dr.

Henry Morton, President of the Stevens Institute of Technology, the visit to both series of shops was made.

On Thursday evening, the guests and members visited in considerable numbers the Fair of the American Institute, to which complimentary tickets had been supplied by the courtesy of its Board of Managers.

On Saturday morning the Inman and International Steamship Company gave the Society the entrée to its fastest vessel, the *City of Paris*. Opportunity was given to see its luxurious passenger appointments, and to visit its engine-room with great thoroughness. A complimentary luncheon was served in the saloon, with toasts and speeches at its close.

CCLVIII.

PRESIDENT'S ADDRESS, 1889.

BY HENRY R. TOWNE, STAMFORD, CONN.
(President 1888-89.)

[Revised from Stenographic Report.]

In opening the twentieth meeting of the Society, and the one which marks the completion of the tenth year of its existence, it is fitting that the first words of the presiding officer should be those of congratulation upon the prosperity and welfare which have marked the development of the Society during the past year. It is surely a source of sincere satisfaction to all of us that the Society has grown during the last year so well and so strongly, not merely in numbers, but in other senses, and in some directions which even more closely concern its usefulness and ultimate strength. There is, fortunately, no law which restricts the president in his choice of the subject to be discussed in his annual address, and availing of that fact, I propose to touch upon three topics of present interest, relating to matters which have occurred during the past year, or which are active in the minds of most of us at the present time. They will be, first, the joint excursion of American Engineers to Europe last summer; second, the contemplated International Exhibition in 1892; and third, a brief résumé of the affairs of the Society.

THE JOINT EUROPEAN EXCURSION.

During the past year there has taken place an occurrence unprecedented in the annals of American engineering societies. A party, consisting of three hundred and sixty, including their guests, went abroad as guests in the first instance of the Institution of Civil Engineers of Great Britain, and subsequently as guests of similar organizations in France and in Germany. It happened that the inception of the excursion came from the Institution of Mechanical Engineers of Great Britain and was extended to this society as being its nearest sister on this side of the ocean. But as soon as the matter became noised in

England, the Institution of Civil Engineers, which is there the parent society of all of their engineering associations, immediately claimed the right and privilege of becoming the hosts of the occasion, and of extending the invitations to each of the three or four engineering societies on this side. It followed from this that the formal invitation came directly to each of our American societies from the Institution of Civil Engineers. The most active part in promoting the project on this side was borne by this Society, but the other societies soon recognized the importance of the matter, and all worked harmoniously together for the common result. It is to be hoped that the temporary union, which thus arose, may be the precursor of another, larger, and more permanent union among our American engineering societies.

The start was made on the 25th of May on the steamer *City of Richmond*, carrying one hundred and seventy members of the party, consisting almost wholly of members of this society and of the American Institute of Mining Engineers. That vessel had been chartered by our party expressly for the occasion and carried no other saloon passengers. The response to the invitation was so large, however, that the overflow, beyond what that steamer could accommodate, necessitated other arrangements, and so another party of one hundred and five followed on the steamer *City of New York* on the the 29th of May. There were still others who came in scattering groups, and some were already on the other side.

The voyage of the *City of Richmond* is a memory which all of us, who had the privilege of taking part in it will ever recall with the greatest pleasure. It was harmonious from beginning to end. A committee was organized on the second day after sailing, and had sessions every day of the voyage—indeed long sessions, because there was much work to be done in preparation for the affairs to be carried out on the other side, more than any of us had realized. The members of the party, both ladies and gentlemen, soon became well acquainted, and the voyage came to resemble a large yachting party rather than an ordinary trip across the Atlantic.

We reached Liverpool on Tuesday, June 4th, and before we had set foot on English soil received a foretaste of English hospitality. There came out to meet us in the Mersey a tender carrying a committee of the local reception committee at Liver-

pool, headed by Mr. Alfred Holt, their chairman (himself reputed to be the largest individual ship-owner in the world), Mr. Daglish, Mr. West, and a number of other gentlemen. They boarded our ship and greeted us with words of hearty welcome, took charge of us during the landing, facilitating our passage of the Customs authorities, and from that time until we left Liverpool were ceaseless in their endeavors for our comfort and enjoyment.

The *City of New York* arrived two days later, in the early morning, and with that day began the regular excursions which had been planned for our entertainment. Our hosts in England were the Institution of Civil Engineers. The individual members taking part in the entertainment, most of whom came expressly to Liverpool to greet us, were the president, Sir John Coode, Sir Frederick Bramwell, Sir Lothian Bell, Sir James Allport, Mr. Adamson, Sir Henry Bessemer, Sir Geo. Bruce, Mr. Cowper, and many others which I will not take time to name; but among them all no name made itself more familiar to us, or will ever be more warmly remembered, than that of the secretary, Mr. James Forrest.

It became necessary for the party, comprising, as it did, members of this Society, of the civil engineers, and of the mining engineers, together with a few members of the electrical engineers, to create some kind of an organization which should represent the united party during its travels in Europe. A joint committee was appointed to accomplish this purpose, and the result of their labors was the selection and recommendation of the following list of officers, who were unanimously elected by the joint party: Mr. Whittemore as honorary chairman, Mr. Henry R. Towne as chairman; and as officers or associates: Mr. Chanute, Mr. Woodbury, Mr. Clarke, Professor Hutton, Mr. Wiley, Mr. Dempster, Mr. Kent, Mr. Archbald, Mr. Baldwin, Mr. Fisher, Mr. Hawkins, Doctor Torrey, Mr. Bond, Mr. Forsyth, Mr. Oberlin Smith, and Mr. D'Invilliers. The treasurer was Mr. Hunt; the honorary secretary, Mr. Emery; the secretary, Mr. Kirchhoff. It is an evidence of the cleverness with which the nominating committee did its work, that out of the twenty-one names above, there are thirteen who are members of the Society of Civil Engineers, thirteen who are members of the Society of Mechanical Engineers, and nine who are members of the Institute of Mining Engineers. The joint committee proved acceptable,

and accomplished its work satisfactorily, although the work proved to be much larger than would have been appreciated beforehand, and demanded a great deal of time and care.

Our first full experience of English hospitality came at Liverpool in the form of a dinner given by Sir John Coode at Liverpool, the evening after the *City of New York* arrived, to a few of the officers of the joint party, followed during the evening by a *conversazione* at the Town Hall, given by the Mayor, Mr. Cookson, and attended by the whole of our party and a great number of ladies and gentlemen from Liverpool—a most brilliant assemblage.

The next morning we divided into two parties, one going to the Mersey Docks under the guidance of officers of the Dock Estate, who have charge of the most vast and expensive system of dock construction in the world, the extent of which is simply marvelous, and to us, in America, utterly unknown. The tides in Liverpool, and indeed all around the English coasts, average nearly thirty feet in height, entirely precluding the use of a wharf system such as we have here, and necessitating the putting of all vessels into docks closed by gates which are only opened for about an hour at high water.

The other party went through the Mersey Tunnel, a great work connecting Liverpool with Birkenhead, which has been recently completed under the guidance of Mr. Rowlandson, the engineer, and then to the Laird ship-yards, where five hundred and seventy-six vessels have been built within the last thirty years. They were entertained at a magnificent luncheon served in a tent on Mr. Laird's grounds, and then visited one of the great steamers for the Hamburg line then being built,—a sister ship of the *Augusta Victoria*,—and finally were brought back to Liverpool, arriving at the great landing stage which is used for tenders and ferry-boats to deliver their passengers upon, said to be the greatest floating structure in the world, and having a total length of two thousand and sixty-three feet.

The next day the party divided again; one section going to Crewe, the location of the great constructive works of the London and Northwestern system, corresponding to Altoona on the Pennsylvania system, where they make steel rails, build locomotives, and conduct most of the mechanical operations of the line. The extent of those works is probably familiar to all of you, but it is interesting to note that the capital of that great

corporation is $528,000,000, with an annual revenue of $51,000,000, and with 60,000 employees. It is also interesting to note that, even in that snug little island, one railway system can control and operate 2,500 miles of line. The Crewe works cover 116 acres of land, of which 36 are under roof.

The other section of the party on that day went to Horwich, on the line of the Lancashire and Yorkshire Railway, and inspected a similar plant there, but one even more interesting than that at Crewe, in this respect,—that while Crewe has grown up almost from the commencement of railway operations in England, and is to some extent a patchwork, although a vast and most highly organized one, the new plant of the Lancashire and Yorkshire Railway, at Horwich, is entirely new, has been built within the last three years, was laid out and organized by commencing with a clean sheet of paper and an unbroken piece of ground admirably chosen, and has a series of vast buildings designed harmoniously, with reference to their intended uses, and in the light of the best and latest modern experience, including that of Crewe. The mechanical engineer of that system, Mr. Aspinall, who has charge of the Horwich works, although a younger man than Mr. Webb, the presiding genius at Crewe, is his equal, apparently, in talent and organizing capacity, and working as he does with this newer and more modern plant, is making a record which certainly will be a good second to that of Crewe. In the manufacturing department, where they make the smaller products, such for example as their switch and signal apparatus, Mr. Aspinall has introduced a great deal of American machinery and American methods of manufacture, and it seemed to me that the place compared favorably with any private establishment I have ever visited. These works cover 85 acres of ground of which 13½ are under roof.

In the evening of that day the two parties united at Manchester, where a reception and banquet were tendered us at the Town Hall, presided over by the Mayor of Manchester, and attended by a great many of the prominent citizens. It was a delightful occasion, and even more elaborate than the reception at Liverpool.

The next day the party visited the great ship canal which is being constructed between Manchester and Liverpool, 35 miles in length, the contract price for which is $28,000,000, where 15,000 men are employed, which is to pass the largest ocean

steamer, and which, although it has been only a short time under construction, is to be completed in January, 1892, and evidently will be.

The next week, being the Whitsuntide Holiday or recess, was utilized for excursions not connected with the engineering part of our visit. The party broke up into two groups, one going through North Wales, the other through the Midland counties and both reuniting in London. It is fitting to remark at this place that all through the trip the courtesies extended to the American engineers by English railway officials were marked and generous to the greatest extent. The London and Northwestern system gave free transportation from Liverpool to London, including a return privilege at whatever time the holder of the ticket desired, and other systems followed later when the party had reached London and made excursions from that point.

On Thursday, the 13th of June, those wonderful eight days of hospitality in London began with a choral service in Westminster Abbey, conducted by Dean Bradley, who gave an address of welcome to our American party; then a brief visit to the Houses of Parliament and in the afternoon a reception by the Institution of Civil Engineers. The latter was opened by an address of welcome from Sir John Coode, the president, the words of which have been beautifully illuminated and framed, and presented to this society, and also to our sister societies here, by the Institution of Civil Engineers, one copy of which is hung in our new rooms. Our party was especially fortunate in having with it at that time one of our oldest and most honored members, to whom was committed the duty of replying to the address of welcome from Sir John Coode, and who did it in a manner which more than fulfilled our expectations. Professor Thurston's admirable address on that occasion was one for which all of our party felt grateful and of which all of us were proud.

In the evening of that day a dinner was given to the party, by the Institution of Civil Engineers, in the old and historic Guild Hall of London, a building which we were told had never before, within memory, been used for any purpose not directly connected with the civic hospitality of the City of London. It was a great compliment. The dinner was elegant beyond easy expression and wa dignified and notable in every particular.

Among the guests of the occasion were the American Minister, Mr. Robert Lincoln, Sir Edward Thornton, Lord Armstrong, Arch-Deacon Farrar, Dean Bradley, Sir Henry Bessemer, Sir William Thompson, Mr. Lattimer Clarke, Sir James Douglass, Mr. Gilchrist, Mr. Mather, Sir E. J. Reed, Professor Unwin, and a great many others whose names are familiar on this side of the Atlantic as well as on the other. One of the pleasing incidents of the evening was the address given by Mr. Lincoln, which was worthy of the occasion and able throughout, and at the close of which he gave utterance to a sentiment, especially complimentary to us, and typical of the character of our times and of the change in sentiment which is taking place in the world. Addressing our united party of engineers, English and American, he said that "engineers throughout the world are doing more than any other agency at the present time, to bring about the brotherhood of the nations, and to render superfluous such offices as that which I now have the honor of holding."

The next day was devoted to visits to the docks and gas-works, to drainage-works, to the great Tower bridge now building across the Thames, to Greenwich, to the Yarrow ship-yard, to various engineering works, and by a fraction of the party, to a visit to Lambeth Palace, where the Archbishop of Canterbury received the guests and conducted them personally through the edifice. On the following day, June 15th, we were taken by special train over the Great Western Railway to Windsor, where the Queen had given special permission for our party to go through the palace, and to see, not only those parts which are usually open to the public, but also the private apartments, which were exceedingly interesting. A small fraction of the party went on that day to the grounds of Mr. Boulton, at Totteridge, where they witnessed a remarkable presentation of the "Midsummer Night's Dream," given in the open air. The evening of the day concluded with a reception tendered to our party by Lord Brassey at his beautiful house in London, where we saw many of the wonderful curios which were collected by himself and the late Lady Brassey during their yachting tours around the world. The following day was a Sunday, and on the next day, Monday, the party went in the morning to see the Royal Palaces in London—St. James's and Buckingham. It was one of the coincidences of this visit that we were greeted there with the strains of Yankee Doodle and Hail Columbia, the day being, as one of our party recalled,

the anniversary of the battle of Bunker Hill. On the afternoon of this day Lady Burdett-Coutts gave a garden-party and reception at her London residence. The following day was devoted to a trip to water-works and pumping-plants, to Hampton Court Palace and Bushy Park, and the day following to similar visits to railway stations and the great plant of the London Electric Supply Corporation, the ladies going to the flower-show of the Royal Botanic Society; and a party of our members, unfortunately a small one, able to avail of the privilege, spent the day in a visit to the residence of Professor Tyndall, who had invited as large a party as his house was capable of entertaining. Those who went received at his hands, I am told, a most cordial and delightful reception, and it is pleasing to mention a fact, which I also learned from those who were fortunate enough to be there, that the response made over the luncheon table to the remarks of Professor Tyndall by our honorary chairman, Mr. Whittemore, were eloquent and beautifully fitting to the occasion. Other hospitalities were extended to individual members of the party on occasions which did not admit of their being made general. One or two of the London clubs gave admission to our members, as had also been done in Liverpool, and in every way the hospitality of our English cousins was cordial beyond any mere words of expression. I think that all of our party, in undergoing these experiences, realized that while there was, of course, a large amount of personal hospitality underlying it, and still more of professional welcome, the true motive prompting these manifestations from our English friends was that of deep and sincere cordiality towards America and Americans. This was made evident to us throughout the whole of our English experience, and it struck me that the feeling of *kinship* on the part of the English toward us is even greater at the present time than the corresponding feeling which we entertain toward them. We Americans look back to England as the mother country, and as such have for it the warmest feeling of affection, but on the part of Englishmen there can be, of course, no corresponding feeling toward this country. The fact of the kinship of the two peoples, however, I believe is more real to Englishmen at the present time than it is even to us, and I think that they realize more clearly than we the fact that together we constitute the two branches of the great Anglo-Saxon, English-speaking race, which, in accomplishment, especially in the industrial world, is at pres-

ent easily the leader among the nations in the march of civilization.

On June 20th our English friends again put us on a magnificent special train, and many of them accompanied us on it, *via* the London, Chatham and Dover Railway, to Dover, and from there by a special steamer across the English Channel to Calais. Our crossing was on a beautiful sunny day, with bright sparkling water, and with no cause for discomfort.

Upon landing on French soil, we again had an immediate greeting of hospitality from our new hosts, represented by members of the French Society of Civil Engineers, who had come from Paris for the purpose. Again we were placed on a special train, tendered by the Northern Railway of France, and taken to St. Omer and Fontinettes, to see a new and unusual canal-lift which had just been completed there, and thence on to Paris. Our hosts, during our French visit, were composed almost entirely of members of the French Society of Civil Engineers, headed by M. Eiffel, the president this year, M. Brull, a past president, M. Contamin, who is the principal engineer of the wonderful machinery palace at the Exhibition, which has recently been awarded, I believe, the prize of 20,000 francs, tendered by an American, for that feature of the Exhibition which, in the opinion of a special committee, appointed to make the award, represented the highest accomplishment and greatest usefulness. The committee's award was to the designers and builders of the wonderful machinery-hall, a building having a span of 330 feet, and a length of about 1,500 feet. The other members of the reception committee were, M. Jousselin, M. Banderali, M. Pontzen, M. Alphand, who is the director-general of the Exhibition; M. Garnier, the world-famous architect; M. Haton de la Goupillière, who is the head of the École des Mines in Paris; M. Gottschalk, M. Charton, and many others. A few members of the joint committee were privileged to be the guests at a small but most delightful dinner given at one of the restaurants in the Exhibition grounds, by a gentleman whose name has been too little associated with this wonderful excursion, Mr. James Dredge, of London, the editor of the journal *Engineering*, and one of the leading representatives of the English section of the late Paris Exhibition. All of our societies are indebted more to Mr. Dredge than to any other one person, for inaugurating the excursion, for enlisting English and French

interest in it, and for contributing to the success of the whole undertaking. One member of our party, the treasurer of this Society, Mr. Wiley, knows the facts, but they are not yet fully appreciated even by the members of our party abroad; and I repeat that no amount of thanks which we can express or tender to Mr. Dredge would cancel the obligation which we owe him.

On Saturday, the 22nd of June, our party went to the Exhibition under the conduct of members of the French Society, and were taken through a portion of it, and then to the Eiffel Tower, after ascending which, we were entertained at a luncheon on the lower platform of the Tower, presided over by M. Eiffel, the president of the French Society, and attended very numerously by members of the French Society and guests, including Mr. Whitelaw Reid, the American Minister, and General Franklin, our Commissioner to the Exhibition.

It would perhaps be encroaching too much on your time to review the Paris Exhibition. The subject is too large to be properly dealt with in the time I could ask of you. It is worthy of note, however, that the chief factor in the success of that Exhibition is its engineering. It is the engineer who has done the most notable things in it; the engineer who has made possible almost every great feature in it; and of course it is the engineer who has been called upon to carry out the plans of the architects and designers. There was great dispute in Paris at first, as to the wisdom of erecting the Eiffel Tower, but that discussion has been closed by the logic of facts. Whether the Tower is, or is not, a thing of beauty, will remain a matter of opinion. Personally, it seemed to me a fitting monument of the great industrial fair, that was there being held. It is certainly graceful, wonderful, and interesting. As to its utility, there can be no question. That has been settled by the fact that it has paid for itself during the period of the Exhibition, and any structure, erected under industrial conditions, may justly be assumed to be successful, if within a reasonable time it appears that it can earn a fair return upon the investment. A further reason, however, for considering it a useful structure is the fact, that to ascend the Tower, and to then behold the wonderful panorama which is spread out before those who do so, is certainly ample compensation to any one for the brief time and little trouble required, and is in itself a justification for the erection of the structure.

Our stay in Paris included many other visits,—to the great sewers, to the Gobelin Tapestry Works, to M. Pasteur's laboratory, to the École des Mines, to the great omnibus and cab companies, to the sewerage and pumping-stations, to Sèvres Porcelain Works, and so on. The social features of our entertainment in Paris included, besides what I have already mentioned, receptions to a part or the whole of our joint party, by President Carnot, by the Prefect of the Seine, and by the Municipal Council. It was a pleasant feature of our reception at the latter place, that one of the speakers on our side, Professor de Garay, of the City of Mexico, a member of the American Society of Civil Engineers, and an accomplished scientist and gentleman, responded most eloquently in the French language, as was also done by other members of our party on other occasions. The Institution of Mechanical Engineers, which had been the first to extend an invitation to us to visit England, happened to have their summer meeting in Paris, just at the close of our stay there, and extended to those of our members who remained, an invitation to their dinner, and to their sessions, so that English hospitality followed us even on French soil. Then came the disbanding of the party, some returning home, others going south, and a considerable number going into Germany, where we afterward heard of them as receiving hospitality even more overwhelming than that which had greeted us either in England or in France. Still others of us came back to London, and a very small number, seven only being obtainable, were privileged to take part in a small but unique entertainment, given by Mr. Dredge, again our host, in order, as he supposed, to enable us to present a handsome silver loving-cup to Mr. James Forrest, as a token of appreciation from the members of the party to him, for what he had done for us during our visit abroad. The committee having the matter in charge, however, appreciated that Mr. Dredge was entitled to a loving-cup as well as Mr. Forrest, and two cups were prepared, each suitably inscribed. Each of the two recipients knew that the other was to be presented with a cup, but neither knew that he was to receive one himself, and there was a very pleasant and amusing *dénouement* when the second cup came out.

On July 22d a number of our party again came together and for the last time accepted the hospitality of the Midland Railway and a special car to Derby, and were the guests there of

Mr. John Noble, the general manager of the company, and several of their directors. After a handsome luncheon in the directors' room, we visited their works, which are similar to those at Crewe, although not quite so large, and ended the day by reaching Liverpool in preparation for the homeward voyage. The party which reassembled in this way at Liverpool numbered more than fifty, and came home together on the *City of Paris*, reaching New York just *three minutes* too late to break the ocean record. The rest of the party came home in scattering groups, but more than fifty of us came together once more in this city on the 25th of August, to be the recipients of hospitalities at the hands of some of our friends at home, at the Engineers' Club, in 29th street, where a handsome dinner was given to the returning guests, a proceeding which was likened to heaping coals of fire on our heads, the hosts being those who had not been able to participate in the pleasures which we had just returned from enjoying. And so ended an excursion remarkable in every sense of the word, absolutely unique and without precedent, and one which will ever be a delightful memory to those whose privilege it was to take part in it.

THE INTERNATIONAL EXHIBITION.

The second topic of which I wish to speak this morning is that of the contemplated International Exhibition in 1892. To be worthy of the occasion and of the people which it is to represent, it must be grand beyond question, an event without precedent in the world. Its chief function will be to exhibit to the nations of the world the industrial accomplishments of a people which, at that time, will number seventy millions, and who, proportionately to their numbers, exceed, in producing efficiency and power, any other people of the world. The exhibition must therefore be worthy of the nation, worthy of the occasion, and worthy of comparison with its predecessors. The latter requirement is one not easily fulfilled in view of what has just been accomplished across the water in that wonderful city of Paris. It may be possible for American engineers and American architects to equal the achievements of the French in the exhibition which they have just closed, but it will be impossible for us to exceed them. We may equal, we may possibly even surpass them, in the mechanical features of the fair, but we cannot hope, in this generation, to rival the accomplishments of the

French in the artistic qualities which are so marked a feature of the Paris Exhibition. It is one of the benefits which should accrue to us from having the exhibition here that it will give further stimulus to the artistic sense in our people, and will help to set on foot, and to give greater vitality to that quality in the development of our arts and industries. International exhibitions are typical of this age. They began, practically, with that of London in 1851, where there were 18 acres under roof, and have been developed since in a series of similar exhibitions, each of which has exceeded its predecessor in extent and beauty. There came next, the Paris Exhibition of 1855, with 29 acres; then London in 1862 with 22 acres; Paris again in 1867 with 38 acres and 10,000,000 of visitors; Vienna of 1873, which is the only exhibition in the series that was not a thorough success; then the Philadelphia Exhibition of 1876, with 48 acres and 8,000,000 of visitors; Paris in 1878 with its 62 acres of buildings and 12,000,000 of visitors; and finally Paris in 1889, with 72 acres of buildings and 25,000,000 of admissions.

The exhibition soon to be held in this country, if held under right conditions, will require an area under roof of 85, 90 or possibly 100 acres. And, again if held under right conditions, will witness an attendance probably reaching 30,000,000, and possibly 40,000,000. These figures are difficult to comprehend· They represent a gigantic undertaking the scope and success of which has not yet been appreciated by those who are discussing its inception and possibilities. It is one of the duties of engineers, in the discussion of the project, that these facts should be clearly brought out and better understood.

It is interesting to consider the probabilities concerning an exhibition to be held here in the near future, in the light of the facts of 1876. The Centennial Exhibition in Philadelphia was held after a long period of commercial and industrial depression, when the country was still using a paper currency and had not resumed specie payments, when our national debt was still a heavy burden, when the country was in many respects still provincial, when our population was 43,000,000 and our railroad mileage about 76,000. The conditions in 1892 will be far different. So far as we can foresee, an exhibition held in 1892 or 1893 will crown a long period of industrial and commercial activity. Specie payments have been resumed and our national debt decreased nearly one-half, the industries of the nation are more

vigorous than ever before, the country vastly developed and far less colonial; our intercourse with foreign nations greatly strengthened, and by that time our railroad mileage will be nearly double what it was in 1876, and our population at least seventy millions. I think it is a fact which admits of no discussion that it is the wish, as it certainly is the interest, of this nation, that the exhibition, when held, shall be international; and here lies the kernel of the discussion as to *where* it shall be held. There is no question but what a great exhibition can be created in Chicago, or in St. Louis, or even in Washington, as well as in New York, but it is a grave question as to whether an exhibition held in any of those places can be made international in the sense that it can be in this city. So far as determined by activity, earnest effort and transparent desire to have the event take place within its limits, Chicago easily claims priority, and all of us should unite in admiring the earnestness, vigor and zeal with which Chicago has set to work to secure the exhibition within her boundaries. But the question should not be considered or determined upon grounds so narrow as those which relate merely to the doings, or the failure to do, of one city or community. The event is certainly national; we want it also to be international. Sectional interest and local pride, still more local jealousies, should therefore be laid aside, and each one who has any interest in the matter, or any opinions, or who can participate in the discussion and settlement of the question, should consider it in the light of a national affair, and should seek that the ultimate decision shall be made in such manner as will best promote the interests of the nation, and not with reference to those of any one locality. As to the selection, viewed in this light, there seems to me to be no room for discussion. In the first place, to foreign peoples New York represents the nation to an unfortunately large extent. The remedy for this will come in time, from greater intercourse between them and us, but it will not be hastened by attempting to hold an exhibition to which we invite foreign nations at a point one thousand miles or more from our seaboard. To do this, on the contrary, will be to erect a barrier which will have the effect of excluding a large proportion of foreign exhibitors and visitors. Our facilities for travel are, we may grant, adequate to carrying from New York to Chicago or St. Louis and back again, without hindrance, all who may come, but that fact is not known to the people on the

other side of the Atlantic, as we know it here, nor can we make it known or make them believe that such is the case. To invite them to come here with their exhibits, and to then travel a thousand miles from our seaboard, will be equivalent to inviting them to stay at home.

The claims of New York as a fitting place in which to hold this exhibition are too numerous to be easily recited, and yet I think it worth while briefly to refer to them. In the first place the site which has been selected is beyond dispute more beautiful by nature, and is already more highly adorned by art, than any in which any other of the great world's fairs has ever been held. I make that statement unqualifiedly and without reserve. No one of these great exhibitions has ever been held at a seaport. We have here, surrounding this island, the most beautiful waterways in the world. The location chosen is so high as to command, from almost any point of it, most beautiful views, but from any elevation artificially created upon that site, most especially from any tower that may be erected, there will be commanded a vista of land and water absolutely without rival in any of the great cities of the world. The facilities for communication between the city and that site, and between the innumerable summer resorts within a radius of twenty-five miles around New York, are such as to give the benefit of hotel and lodging accommodation already in existence which is probably adequate to any demand which would be put upon it, but which is capable of easy and indefinite extension before that time shall come. In addition, however, is an immense facility for still adding to these accommodations by resort to floating hotels, utilizing for that purpose the river and harbor boats which will be required to carry people to and from points along the various water fronts. The facility for reaching New York from the interior of the country is greater than that of any other one of our seaboard cities. The facility for reaching it from Europe is absolutely unrivaled. All of the other cities combined do not possess shipping communications at all equal to those already established from New York. It is not a correct index of the suitableness of a locality to strike a long radius from its center and to count the population within the circle described, if in so doing the same rule be applied to this city and its facilities on the eastward, or its water communications, be ignored. An attempt has been made in that way to demonstrate that a greater

population is within easy reach of certain inland cities, but in so doing the fact has been ignored that from New York's eastern front, which represents no population, stretch forth the immense and constantly increasing lines of communication which link together the New World and the Old.

There are many current events which emphasize the importance to the country of holding the proposed exhibition; the Pan-American Congress, now in session, the avowed object of which is to foster closer relations with our neighbors to the south, the creation of our new navy, the interest manifested throughout the Union that our mercantile marine should be resuscitated, the desire which all of our manufacturers evidence, that the markets open to their products should be enlarged and extended beyond the boundaries of the country, and the fact, which both of our political parties have plainly recognized, that we are outgrowing our industrial childhood and are rapidly approaching a point where protection, which has done so much to foster our industries, is no longer needed to the same extent as in the past, a recognition of which fact will, in the near future, enable us to enter in competition for the markets of the world on better terms than we have ever done before. All of these facts illustrate and emphasize the desirability to this people of coming into closer communication with the nations on the other side of the Atlantic. The Centennial of 1876 was a great blessing to this country as an educator of our own people, and the wonderful development of the artistic element in our own industries which resulted from it has been beneficial to an extent not easily exaggerated. The value of the exhibition now to be held should be of a different kind, however. While it also will help to educate us in a similar manner, I believe that its greater value will be in educating the people of other countries to a better knowledge of the industrial accomplishments of this nation, and to the fact that the United States has become one of the buying markets of the world, one in which the buyers of all countries are interested and concerned, and of which they need, in their own interests, far better and larger knowledge than they have at the present time. In no way can we put that knowledge before them better, and interest them more in understanding it, than by holding an exhibition which, as I have said before, shall be international, and which shall attract to it the largest possible number of visitors from the other countries of the world.

But how is such an exhibition to be created? and when can it be held? The Paris Exhibition of the present year had at its head M. Alphand, who has been identified with each of the preceding great exhibitions there, and who has for forty years been intrusted with the management of the great parks and the constantly recurring *fêtes* of Paris. He had associated with him M. Berger, who had been the administrator of six preceding expositions. Its art exhibit was in charge of M. Méry, who was in charge of the great Paris Salon for many years. Its architectural features were largely directed by M. Garnier, the architect of the Paris Opera House, a man of world-wide reputation, and its engineering department had in its leadership such men as Eiffel, Contamin, and Charton. Even with the leadership and help of such men as these, and of an army of them—the list of officials of the Paris Exhibition covers many pages of a printed book—even under such leadership as this, how long did it take France, with all her resources and all her readiness for such work, to accomplish the exhibition which has just been closed? Four and one-half years. That is the length of time which passed from the enactment of the first law decreeing that the exhibition should be held until its accomplishment; fifty-four months, to state it otherwise. It is interesting to note other figures relating to the time required. The first credit of 100,000 francs, and the first advertisement for designs for buildings were issued forty-five months in advance of the date of opening. The financial scheme was promulgated, full-fledged and complete, and the executive staff organized, thirty-four months in advance, and after all of this long period of preparation, under the leadership of the ablest men of the nation, the time required for actual construction was thirty months. We have now remaining before the first of May, 1892, twenty-nine months. *It is the duty of the American engineer at the present time, in connection with this enterprise, to say to the people of the nation that it cannot be done in* 1892; *we must take more time.* It is possible, beyond question, for New York or Chicago, or any of our great cities, to hold an exhibition in 1892, and to make it huge and showy, heterogeneous and discreditable; but that it can be done in any manner worthy of the nation and of the occasion, in the short period of twenty-nine months, and especially that it should have grafted into it any reasonable amount of artistic excellence, which will make us not ashamed to have it compared with what has been accom-

plished at Paris, is, in my opinion, simply impossible. And if I am right in that view of it, I beg to suggest to the members of the profession that, in discussing the subject everywhere, this fact should be stated, together with the reasons on which it is based, as soon and as fully as possible. Sentiment is the only reason for attempting to have the exhibition in 1892. In every other respect it would be better if held later. And surely sentiment in this connection should be made secondary to success.

To indicate the possible benefit of such an exhibition, not merely to the locality in which it takes place, but to the nation at large, I have put together a few of the many statistics relating to the Paris Exhibition of 1889. The cost of admission in Paris averaged about ten cents; it was more at first, and less at the close of the exhibition. Ten cents in Paris represents easily thirty cents in America, taking into account the difference in rate of wages and the different values of money, and especially the fact that our working people are better off, far more ready to spend money for such things, and far more accustomed to travel than the corresponding classes in France. The average attendance at the Paris Exhibition was 125,000 per day, and on Sundays 300,000. The maximum on any one day was 405,000 persons. The total attendance was 25,000,000. The financial facts, briefly, were that the original estimates, made with the most wonderful comprehensiveness, intended to outline in advance every detail of receipt and expenditure, to appropriate a definite amount for each item of construction and expense, and to rigidly hold the administration to expenditures within those limits,—those estimates were originally forty-three millions of francs, and were subsequently increased to forty-six and a half millions of francs. But the enormous attendance, especially during the latter part of the exhibition, increased the receipts some eight millions of francs, nearly all of which is profit, which amount will be divided among the contributors to the initial fund. The Eiffel Tower cost six and a half millions of francs to build, and its net receipts during the exhibition happened to be just the same amount, leaving the property fully paid for, in the ownership of the present company, with a concession for twenty years, during which time the profits will revert to them. The railways entering Paris carried, during the exhibition, five millions more passengers than they did in the corresponding period of 1888, an increase of fourteen per cent. The *octroi*, or local

duty on merchandise carried into the city for the first nine months of 1889, was greater by two millions of dollars than during the previous year, a direct return to the city for its contribution to the cost of the exhibition buildings. The omnibuses and tramways carried fifteen millions more passengers than in 1888. The river boats carried twenty millions more, or double the number of the previous year. The Circular Railway around the city carried thirty thousand people per day more than in 1888, and the net earnings of the six leading French railways increased during the exhibition period, as compared with the year before, \$10,900,000. It is estimated that of the visitors to the city of Paris five millions were French people, and six and one-half millions foreigners, the latter being much the more liberal spenders. One authority estimates that the money expended by visitors in the city, French and foreign, was \$358,000,000; but another and more conservative authority, which I prefer to accept, places the amount at \$250,000,000. Either of these amounts represents an enormous sum. There were sixty thousand exhibitors; there were seventy congresses held in Paris during the summer; and the whole affair was most truly a national glory.

The American engineer, as I have told you, has, at the present time, in my opinion, a first duty in reference to the contemplated exhibition; that is, to tell his countrymen that it cannot be properly held in 1892, and that we should have at least one more year in which to make preparation for it, and this without any reference to the locality in which it is held. The engineer, whenever it shall be held, must bear the chief burden of preparation and construction, and the profession at large, civil, mechanical, electrical especially, and even mining engineers, are all concerned to see that this exhibition, when it takes place, shall be made worthy of the nation. The work, of which they will each have to do their share in the preparation, planning and construction, will be large and profitable to them, both in the immediate employment it will afford, and still more in the professional experience which it will afford, and in the professional intercourse it will bring with brother engineers from other parts of the world. And in this connection we must remember, especially those of us whose privilege it was to take part in the excursion of last summer, that one of the obligations which will come with the exhibition, when it is held, will be to act as hosts

to our English, our French, and our German friends, who, we then hope, will reciprocate the visit which we have made to them this year.

THE SOCIETY.

The last subject to which I will refer, and which I can treat more briefly, relates to the affairs of our own society. This, the younger in the sisterhood of American engineering societies, had its conception in February, 1880, at an informal meeting of which the lamented A. L. Holley was chairman; and at the risk of encroaching more than is reasonable upon your time, I cannot forbear quoting a few sentiments from the address which he made at that time, which seem to me to be especially fitting and interesting in the light of events which have occurred between then and now. He said:

"It seems very remarkable that an institution of mechanical engineering—which underlies all engineering—has not long ago been organized in this country of mechanical engineers. I confess that in thinking over the range of mechanical engineering, with reference to our proposed society, I was astonished at its magnitude. I had never realized it before. . . . .

"I would not underrate—I could not too highly magnify—the wide and profound scientific knowledge employed in locating and planning these vast works of civil engineering—the canal, the harbor, the railway, the tunnel, the pier, the breakwater—I only emphasize the fact that our own profession of mechanics and dynamics underlies their construction and utilization—it is the intermediate power between nature on the one hand, and the artificial structure and the artificial work done on the other hand.

"Should the architect and the civil engineer say that the mere molding and assembling of members is not worthy of a professional name and *status;* the mechanical engineer may reply that the mere calculation of strains from well-known formulæ, and the mere grouping of conventional forms is no more worthy. The genius that reaches the harmony of perfect construction and perfect beauty (which are interchangeable terms) in Nature's inert materials, may not be loftier than that which as perfectly utilizes and governs her wild and capricious forces.

"We should also briefly consider the advantages and character of our proposed organization.

"*First.* The most obvious advantage is the collection and diffusion of definite and much needed information, by means of papers and discussions.

"*Second.* A less obvious, but it seems to me, a more important advantage of organization, is the general personal acquaintance thus promoted, among engineers and the business men associated with them. It is being found out that fifty men can impart more information to one man than the one can impart to the fifty.

"*Third.* The habit of writing and discussing technical papers is of very great importance. It engenders habits of thought, at once rapid and accurate. Any man can work better who can formulate the merits and defects of work. . . .

"Finally, the rapid and healthy growth of the society will largely depend on the character of the first few years' papers. With men of work and of note for early officers, and a goodly number of really important and well-written papers at the start, the success of the Society of Mechanical Engineers is assured."

Holley's prediction has been fulfilled, but it devolves upon us to carry on the work, if we can, to do even better in the future than in the past

On November 5th, 1880, the first regular meeting of the Society took place under the presidency of Professor Thurston, with a membership then of 189. At the same time the membership of the Society of Civil Engineers was 611, and that of the Institute of Mining Engineers, 843. All have prospered in the interval, but none more so than we. Our membership to-day is 1,048, that of the Civil Engineers, 1,280, and that of the Mining Engineers, who require no professional qualification, 1,986. The aggregate of all three Societies is something over 4,000, or with the new and younger Society of Electrical Engineers, probably 4,500. It is interesting to note that the membership of the Institution of Civil Engineers of Great Britain is 5,728, and that their increase for last year was 198, ours being 173.

During the year which has just closed the Society has moved into new offices, at No. 64 Madison Avenue, far more suited to its work and efficiency than any it has yet occupied, a change which will undoubtedly increase its usefulness, especially as to the library. Our meetings are held, however, not in the Society's rooms, nor always in this city, but at points scattered throughout the country; and in my opinion that policy should be rigorously adhered to, and our meetings held constantly at new points in order that the membership in all parts of the country may have the privilege of attending them, and that our membership may be increased in all directions and thus be thoroughly national and not local. Local organizations exist already, and are to be encouraged as affording facility for frequent meetings and the discussion of matters of local interest, but it is to be hoped that ultimately they may become chapters of the larger society. And in this connection I wish briefly to refer to the fact which was mentioned more than once during our European trip of last summer, and which seems to be striking deeper and deeper root into the minds of the members of all of our societies, namely, that the time is approaching when it will be both desirable and possible to create some new organization which, without in any

way interfering with or diminishing the autonomy and usefulness and individuality of the existing societies, shall unite in one larger and more thoroughly national organization the older and leading members of each ; a society which shall include in its ranks the best representatives of American engineering, and of which membership will be a valued privilege and one which will convey distinctly an understanding of high professional character on the part of those who are in its ranks. Especially is this desirable with reference to its bearing upon our relations to engineers from other countries, and for this reason it is to be hoped that some such organization may be accomplished before the proposed international exhibition, when we hope and believe that large numbers of foreign engineers will come to this country as the guests of American engineers.

Finally, as to the Society, it seems to me that notwithstanding the success which has been achieved in the past, and the large fulfillment we have made of Holley's prediction and hope, there is still room for further improvement, especially in our methods of business, in the care with which we edit our transactions and publications, and in the providing of means and facilities for undertaking special investigations, as is done by the English societies, and as cannot often be done by private individuals, and for which purpose funds should be provided, for the building up of our library (a matter which will be touched upon by the Library Committee in its report later), and in general for stimulating and increasing the usefulness of this society of engineers. All of these are among the duties devolving upon the members, far more than the mere effort to increase our numbers, which are growing as rapidly as is healthful, and we cannot do better, all of us who have the interest of the Society at heart, than to remember that in such ways, and especially in the effort to elevate the character of our transactions and to contribute papers covering original investigations and matters of professional value, we will most efficiently bear our share in bringing about the result which we all so earnestly desire, namely, the further growth in prosperity and usefulness of the Society of which we are members.

CCLIX.

*TABLE OF THE PROPERTIES OF STEAM—THEIR USE IN STUDY OF STEAM-ENGINE EXPERIMENTS.**

BY V. DWELSHAUVERS-DERY, LIEGE, BELGIUM.
(Honorary Member of the Society.)

INTRODUCTION.

M. V. Dwelshauvers-Dery, the distinguished head of the School of Mines, at Liège, Belgium, has recently given much attention to the study of the exchanges of heat occurring in the cylinder of the steam-engine, resulting in losses of energy of serious amounts, as illustrated by the experiments of Watt, of Clarke, of Hirn, and of Isherwood. In the course of this work, he has devised an exceedingly interesting and useful method of exhibiting the process of exchange occurring in any given case in which the data have been made available by experiment, and has given graphical constructions representing this action and its effects. He has also, recently, prepared a set of steam tables to be used in the course of such work; which tables he has kindly consented to lay before the American Society of Mechanical Engineers. These tables are here presented, the measures being given in British units, and in such form as may best facilitate the calculations to be made from data obtained in formal trials of the steam-engine.

The text accompanying the tables embodies a succinct exposition of the theory of M. Hirn, whose views and methods are closely followed by the author of the tables, and two examples are given, illustrating the method of M. Dwelshauvers. Appended, also, is a plate in which is exhibited the complete graphical construction of the process and results, as devised by the distinguished author, our colleague, and showing the quantity of heat, and of equivalent work produced; the quantity lost by the cylinder walls, and the amount of the remain-

* Translated from the French in the office of the Society.

ing losses. The total of these several quantities measures the amount of heat and of energy supplied to the engine. This "diagram of exchanges," as its author calls it, may, also, not improbably, prove to be of very great value to the engineer seeking to determine, in advance of the construction, the probable demand of a proposed engine doing a specified quantity of work. The problem of discovering the methods and the laws of wastes occurring through the interaction of the steam and the enclosing surfaces of the steam cylinder is, to-day, the one important problem in the theory of the engine. This problem, attacked mathematically by Grashof, and experimentally by a dozen well known engineers, is exhibited and illustrated by the graphical work of Dwelshauvers in such manner as is likely to prove of real assistance in promoting the final complete solution. The author of the paper and of these tables has been a very old friend of Hirn, whose work has been familiar to him from its inception and has been closely followed from the beginning. His papers have done much to make useful the work of the great physicist and engineer who has given us so much valuable data, and this process of treatment is not the least important of his accomplishments.

Those who are specially interested in this subject can get still more complete ideas of the method and of the results of its application by studying the paper of M. Dwelshauvers-Dery recently issued by the British Institution of Civil Engineers (No. 2403 A.) on the trials of Mr. Bryan Donkin's engine. It will be seen that the method consists in the application of the processes here indicated to the computation, from the measurement of the indicator-diagram, of the quantities of heat lost by the steam in the engine in the three principal directions of expenditure: transformation into mechanical energy; waste by way of the metallic surfaces of the interior of the cylinder; thermodynamic and thermal losses in other directions. By this method of study of the work of Mr. Donkin, it is found that the jacket does not always produce a flow of heat into the engine, a result probably suspected by every expert observer and experienced engineer having much acquaintance with the steam-engine, but probably never before so completely proved. The fact accounts for the very diverse testimony often given respecting the action and efficiency of the jacket. The process pursued by our author enables us to divide the engine into its two

parts, treating the two ends of the cylinder separately, and discovering, through the examination of their differences, many facts not otherwise obtainable. A distinction never before quantitatively made is that of the positive and the negative flow through the surfaces receiving heat, and through those restoring heat. In some cases, the one area and quantity is the greater, in others the other. Thus the jacket may sometimes produce marked economy, while in other cases it may produce none or may even cause loss. We find, also, that the jacket may produce economy by simply preventing external losses, giving absolutely no heat to the steam, but simply preventing its losing as much as it otherwise would, at the critical instants of transfer to the metal of the cylinder. It is easy to show, by this or other methods, that the use of the jacket is ordinarily advantageous by preventing transfer of heat to the metal of the cylinder during admission, and that the function of the jacket is usually substantially completed at the close of this period, and, consequently, that the engine of large diameter and small stroke, a given volume being demanded, and with a jacket on its heads, has, ideally at least, an advantage, economically, a fact which has probably been suggested by many familiar data to many engineers.

These comments are presented to the society in the hope that the work of the distinguished author of these tables may become more familiar to members generally, and that they may become useful to all who are engaged in the study and investigation of the efficiencies of the steam and other heat engines. The writer esteems it a pleasure and an honor to have been permitted to make this presentation.

R. H. THURSTON.

PROPERTIES OF STEAM.

1. It would, perhaps, be useful to those who are studying the steam-engine, to present to them tables of saturated steam more complete than they are ordinarily made. Such tables are in existence in French units, and they have been translated into the British equivalents, making use of the fundamental data taken from most reliable authorities. No attempt has been made to obtain a precision beyond dispute. The attempt has not been to make a learned disquisition, but simply a useful

paper. The author will feel entirely satisfied if this end is attained.

2. It will be supposed that the reader is familiar with the properties of dry and saturated steam as well as those of water under corresponding pressure at a given temperature. It will be sufficient, then, to recall them succinctly: When steam is saturated under a given pressure, it has a temperature and a density which are known. For example, steam at sixty pounds pressure per square inch has a temperature of 292°.52 F., and a pound of that vapor occupies a volume of 7.0328 cubic feet; or again, a cubic foot of that steam weighs 0.14219 lbs. These figures are given in the table in the following order:

The first column gives the pressure of the steam, p, in pounds per square inch.

The second column gives the same pressure, P, in pounds per square foot; that is to say, $P = 144\,p$.

The third column gives the temperature in degrees Fahrenheit.

The fourth column gives the volume of a pound of saturated dry steam expressed in cubic feet.

The fifth column gives the weight of a cubic foot of dry saturated steam at the pressure given in the first column and expressed in pounds.

3. The five following columns give in British thermal units the different quantities of heat to which the following names are given:

q, heat of water under the pressure, p, of the first column and at the corresponding temperature, t, of the third column. It is the number of British thermal units contained in a pound of water at the temperature, t, and the pressure, p, taken above that amount of heat which is contained in water at the temperature of melting ice; *i. e.* + 32° F. From this definition, if the number of thermal units is sought which is necessary to raise the temperature of a pound of water from 102°.09 to 233°.02, look out in the sixth column the values of q which correspond. That is, $q_0 = 70.09$ and $q_1 = 202.32$, and the difference $q_1 - q_0 = 132.13$ thermal units will be the required answer.

ρ, *interior latent heat of a pound of dry saturated steam:* that is to say, the heat which corresponds to the work necessary solely to overcome the internal molecular forces which hold the molecules at the distances at which they must be while the mass is in a liquid state, in order to bring them to the distances at

which they will be found when the mass is in the condition of saturated steam.

$A\ P\ u$ is the *exterior latent heat* of a pound of steam generated under the constant pressure P pounds per square foot, while the piston sweeps through the volume, u. This demands explanation. The exterior latent heat has been measured under the following conditions: a pound of water is put in a cylindrical vessel at a temperature t, and in the cylinder is a piston which exerts on the surface of the water a pressure of p pounds per square foot, corresponding to t degrees for saturated steam. To give precision to the discussion, let $t = 292^\circ.52$. The table gives $P = 8{,}640$ pounds per square foot. In this condition, the pound of water occupies a volume, σ. Heat is furnished in a measured quantity. Then the piston rises in the cylinder, the temperature of the water remains unchanged, but the water evaporates little by little until there is nothing but dry and saturated steam at the pressure P, and the temperature t. At this instant, stopping the experiment, the volume u, traversed by the piston, is measured and expressed in cubic feet. Hence follows naturally $\sigma + u = v$. Where v, as above, denotes the volume occupied by the pound of saturated steam at the pressure P, experiment shows that $P = 8{,}640$ pounds per square foot, and $t = 292^\circ.52$.

$\sigma = 0.0160$; $u = 7.0168$ cubic feet.

Whence we conclude that the heat furnished has done a work $P\ u$ foot lbs. to raise the piston against the exterior pressure, and since each thermal unit corresponds to 772 foot pounds it follows that the heat which disappeared by doing this exterior work is denoted in thermal units by $\frac{P\ u}{772}$, which becomes $A\ P\ u$ if $A = \frac{1}{772}$.

r is the *total latent heat* of a pound of dry saturated steam; that is to say, it is the sum of the two preceding quantities,

$$r = \rho + A\ P\ u.$$

It is the total heat which disappears in the operation of vaporization under constant pressure, concerning which reference has just been made; heat which has become latent since the temperature has remained constant as well as the pressure.

Finally λ is the *total heat* of a pound of dry saturated steam. It includes the heat of the water and the latent heat of the steam formed under a constant pressure.

From this follows:

$$\lambda = q + r = q + \rho + A P u.$$

If then a pound of water is taken at 32 degrees, and is exposed to constant pressure under a piston amounting to P pounds per square foot, and if heat is furnished in measurable quantities, the temperature of the water will at first increase until t corresponds to P, and the piston will only have moved a very little; afterwards the temperature will remain stationary and the piston will rise until all the water has become steam. In this complete operation there has been expended a quantity of heat which is known by the name of total heat, and is ordinarily designated by the letter λ.

4. If now the above operation was carried further, the temperature of the steam would rise at the same time that the piston continued to rise, and there would be expended for each degree of increase of temperature per pound of steam, a quantity of heat C_p, to which the name is given of specific heat at constant pressure. This quantity varies with the pressure, but in practical applications no risk is run of sensible error, if it is made equal to 0.48 thermal units for steam gas. If λ' is called the total heat of a pound of superheated steam at t' degrees, and which would be saturated at t degrees,

$$\lambda' = \lambda + 0.48\,(t' - t) \quad . \quad . \quad . \quad . \quad . \quad (1.)$$

5. *Mixture of saturated water and steam.*—In steam-engines dry and saturated steam may be said never to occur. It is nearly always wet or mixed with water, and rarely superheated. In order that a mixture of water and steam may be well defined, its composition or percentage of dryness must be given in addition to the pressure. To make this matter more lucid, suppose a mixture of an entire weight of M lbs., and which contains m lbs. of pure saturated steam, and therefore $M - m$ lbs. of water at the pressure p and the temperature t given in the tables as corresponding. As soon as p is given, the tables give at once all the quantities, P, t, v, d, q, ρ, APu, r, λ. But the ratio $\frac{m}{M}$, that

is to say, the percentage of dryness which is ordinarily represented by the letter x, is independent of these quantities, and must be one of the experimental data, or else amongst the data must be the two quantities M and m.

6. The name *of internal heat of a mixture* of saturated water and steam is given to the heat contained in this mixture above that contained in the same weight of water at 32° F., whatever may be the procedure by which the mixture has been raised to its actual condition. We will designate this quantity of heat by U. It is made up, obviously, of the heat of the water, Mq, and of the external heat of the steam, $m\rho$. Hence:

$$U = Mq + m\rho \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2.)$$

The internal heat of M lbs. of superheated steam, occupying a volume V cubic feet under a pressure of P lbs. to the square foot, is given by an empirical formula:

$$U' = 857M + 0.003872PV \quad . \quad . \quad . \quad . \quad (3.)$$

7. When a mixture of water and steam is acting in the cylinder of a steam-engine, if its total weight remains constant, its composition and temperature are varying at each instant, and for two reasons: there is heat required for the exterior work, and there is heat transferred to and from the metal. Every theory which does not take account of this last phenomenon is fallacious and useless. The practical theory which we hope to present in order to show the use of the tables has been made to take account of this action. It should be remarked, at the outset, that the internal heat of the mixture at work in the steam cylinders must vary at each instant, and according to an obvious law. Under the conditions existing after the points of cut-off in an engine, the mixture at the beginning contains U_0 thermal units, and at the end there is only U_1 thermal units, U_1 being less than U_0. This is because it has given up T_a thermal units to do the exterior work, and R_a thermal units to heat the metal of the cylinder. There must, therefore, necessarily exist the relation $U_0 - U_1 = T_a + R_a$.

And whatever be the elements designated by 0 and 1, the same conditions will exist unless, during the period considered, new steam gets into the cylinder, bringing additional thermal units, or unless some steam leaks out, carrying with it certain ther-

mal units. In these cases it will be necessary to take account in the equation of the units brought in or removed.

8. *Practical theory of the steam-engine.*—The object of this theory is to study the steam-engine as it is, that is to say, to explain the phenomenon met with in an engine trial. The general principles will be brought out, but there will always be in mind a single cylinder engine rather than a compound.

9. *Working Regime.*—A calorimetric experiment on a steam-engine demands that it should be running at its regular rate, that its velocity should be constant, that the pressure of steam, the point of cut-off, of compression, the back pressure, the lead, etc., etc., and, in fact, all these quantities should be constant. Since this ideal is impossible to attain, it should be reached as closely as possible and the means calculated, which will be sufficiently exact when the length of the test is long enough. In what follows it will be supposed that the two strokes of the piston, which constitute a revolution, are identical in all respects, so that but one stroke will be considered.

10. *The perfect regime necessarily presupposes that the steam and the metal are, at the end of the stroke of the piston, identically in the same thermal conditions as at the beginning.* It follows that at each stroke of the piston, the sum of the quantities of heat which the steam receives in the cylinder is equal to the sum of the quantities of heat which it loses there; and the sum of the quantities of heat which the metal receives is equal to the sum of the quantities of heat which it loses. These fundamental principles are general, but the application that we shall make of them has particular reference to single cylinder engines. Experience must give the necessary figures to determine all these quantities of heat thus brought into the cylinder with the steam, or carried off to the condenser with the steam, those which disappear to do the exterior work, those which the metal has extracted from the steam to pass outside by radiation, or those which a jacket would have furnished to the steam through the metal, the heat which is found in the steam remaining in the cylinder at the moment where cut-off occurs, and where the compression begins, etc., etc. The calculation and plotting of the figures given by a trial have for their end to add the diagram of exchanges of heat to the diagram of pressures.

11. Here are the data on this subject, which experiment must furnish.

An indicator diagram, completed by marking on a proper scale the volume of the clearance, v, measured experimentally on the machine itself. The illustration hereafter shows the quantities which the diagram and the special data of the machine ought to furnish. It is not by accident that the atmospheric line has not

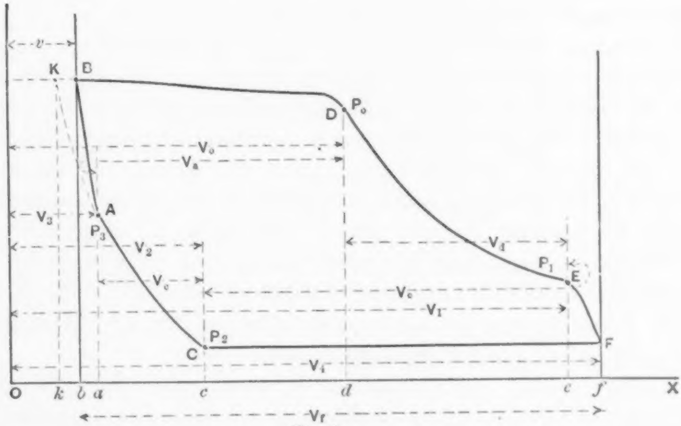


Fig. 3.

been indicated, since it is only of use to obtain the line of vacuum, OX, which is drawn below the first at a distance which, on the scale of the diagram, represents the barometric pressure existing during the trial.

12. Measures taken directly from the machine have given the following volumes, expressed in cubic feet:

v, volume of the clearance space.
V_0, " occupied by the steam at the point of cut-off.
V_1, " " " " " end of expansion.
V_2, " " " " " " exhaust.
V_3, " " " " " " the compression.
V_4, " " " " " " " stroke.

$V_a = (V_3 - v) + (V_0 - v)$ is the volume swept through during admission.

$V_d = V_1 - V_0$ is the volume swept through during expansion.

$V_e = (V_4 - V_1) + (V_4 - V_2)$ is the volume swept through during exhaust.

$V_c = V_2 - V_3$ is the volume swept through during the compression.

$V_f = V_4 - v$ is the volume swept through during a stroke.

On the diagram the following pressures are measured in lbs. per square foot:

P_0, pressure at the end of admission.
P_1, " " " expansion.
P_2, " " " exhaust.
P_3, " " " the compression,

as well as pressure and back pressure at any moment, so as to enable to calculate the areas of the diagram, or the work corresponding to these areas in foot pounds, reducible to thermal units by dividing by 772.

$T_a'' =$ area $bBAab$ is the work in thermal units during the admission before the beginning of the stroke.

$T_a' =$ area $bBDdb$ is the work in thermal units during the admission forward stroke.

$T_a = T_a'' - T_a'$ is the work in thermal units during the admission.

$T_d =$ area $dDEed$ is the work in thermal units during the expansion.

$T_e' =$ area $eEFfe$ is the work in thermal units during the exhaust before the end of the forward stroke.

$T_e'' =$ area $fFCcf$ is the work in thermal units during the exhaust, backward stroke.

$T_e = T_e'' - T_e'$ is the work in thermal units during the exhaust.

$T_c =$ area $cCAac$ is the work in thermal units during the compression.

$T_f =$ area $bBDEFfb = T_a' + T_d + T_e'$ is the work in thermal units during the forward stroke.

$T_n =$ area $fFCABbf = T_e'' + T_c + T_a''$ is the work in thermal units during the backward stroke.

$T = T_f - T_n = ABDEFCA$ is the indicated work in thermal units corresponding to a stroke of the piston.

13. *Note:* The steam has done a work which is not indicated on the diagram; it is that which is represented by the area $aAKka$ and which is necessary to accomplish the compression of the steam in the clearance in order to give it a pressure equal to that of the steam coming in the cylinder. Because the magnitude of this work is not known, it will be counted in the

heat exchanged between the steam and the metal during the admission.

14. There is another thing upon which an uncertainty exists, and that is the composition of the mixture of steam and water which remains within the cylinder at the moment when exhaust ceases and compression begins. M. Hirn has triumphantly shown that in his experiments and in general it must be supposed that at this instant the mixture contains nothing but steam dry and pure, all the water which covered the walls having been vaporized and expelled into the condenser during the exhaust. This will be admitted, and thence our calculation can ascertain the weight M_c lbs. of steam acting during the compression. In fact the volume V_2 of steam in cubic feet can be ascertained from the diagram, and its pressure P_2 at this instant. From the tables the value of δ_2 is deduced which is the weight in pounds per cubic foot, and the expression will follow:

$$M_c = V_2 \delta_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

15. Experiment must further give the following data: the weight M_a pounds of steam which passes into the cylinder at each stroke of the piston and its degree of dryness x or the weight m of pure steam which is contained therein as well as the boiler pressure. This datum will serve to calculate the heat which the steam brings with it for each stroke of the piston in the cylinder. It will be called Q thermal units. From what precedes, its value will be

$$Q = m\,\lambda + (M - m)\,q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

In this expression λ and q are given by the tables at the line which corresponds to the pressure of steam in the boiler.

16. During the expansion, the weight of the mixture in action in the cylinder is, therefore,

$$(M_a + M_c),$$

and during the compression, M_c.

Hence if the steam is saturated its internal heat will be as follows: during the expansion

$$U = (M_a + M_c)\,q + m\rho \,.\quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

During the compression

$$U = M_c q + m\,\rho.$$

If the steam is superheated make use of formula 3.

17. If there is a steam jacket the water which comes from the condensation is weighed after it has been cooled, and its weight will be ascertained per stroke of piston. Let it be called M_j lbs. This steam is condensed under the mean pressure of the boiler, which is ascertained by trial and by calculation. For each pound the jacket will have furnished r thermal units given in the table opposite the pressure. The jacket has then furnished Q' thermal units, and

$$Q' = M_j\, r. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

If the jacket is filled up with anything other than steam, the heat brought in will be evaluated in an appropriate manner per piston stroke, and will still be called Q'.

18. Part of the heat brought in by the jacket will have penetrated through the metal and reached the steam; another part will have gone outside and is lost by radiation. The radiation per piston stroke should be evaluated experimentally. For example: introducing steam into the jacket, or, if there is no jacket, into the cylinder while the machine is at rest, and keeping it there during a sufficient interval, knowing afterwards the duration of a piston stroke during the trial which is studied, the quantity of heat lost by radiation can be deduced therefrom for each stroke. It will be called E thermal units.

19. When the engine is condensing, the weight of water is measured which leaves the condenser, and from this is deduced the weight of cold water M_e lbs. which has been introduced into the condenser for each stroke. Its initial temperature t_i is measured and its final temperature t_f. The tables give the heats of the water q_i and q_f which correspond. Hence the heat rejected per piston stroke by the cold water is ascertained C thermal units by means of the equation

$$C = M_e\,(q_f - q_i). \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

20. Finally in the condenser is the weight M_a lbs. of water which comes from the condensed steam and which is at the temperature t_f if the condensation is effectuated by injection, and at the temperature t_f' if by a surface condenser. It follows that a second part of the heat rejected into the condenser, which may be called c thermal units, will be determined by one of these two formulæ,

$$c = M_a\, q_f \text{ or } c = M_a\, q_f'. \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The heat rejected in the condenser will be the sum or

$$(C + c). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

When there is no condensation of the steam this heat rejected in the atmosphere cannot be evaluated direct, but the same name will be kept for it and the same notation.

21. From what precedes, the vapor which works in the cylinder carries there Q thermal units at each piston stroke. It receives from the jacket $Q' - E$ thermal units. It loses T thermal units to overcome the exterior work, and it carries with itself into the condenser $(C + c)$ thermal units. As the *regime* is reached the sum of all these quantities of heat is zero. (See paragraph No. 10.)

Hence follows the first fundamental equation of the practical theory:

$$Q + Q' - E - T - (C + c) = o$$

or

$$Q + Q' = T + (C + c) + E. \quad . \quad . \quad . \quad . \quad . \quad (I)$$

When there is condensation all the quantities which enter into this equation are given experimentally. This equation can only serve in this case as a check. If the second number should differ too much from the first it would mean that the trial had been badly conducted. Without condensation this equation can serve to determine the value of the rejected heat $(C + c)$, but the means of checking is not direct.

22. Because the engine is in working *regime* the metallic walls of the cylinder reach always the same temperature at the end of the cycle that they had at the beginning. The conclusion is that the sum of the quantities of heat received and given out by the metal at each stroke is equal to zero. The quantities of heat received or given up by the metal, or, in other words, exchanged between the metal and the steam, will be designated in a general way by the letter R when they are expressed in thermal units. The subscript indicates the phase during which the exchange is measured. Hence:

R_a thermal units is the quantity of heat exchanged between the metal and the steam during admission.

R_d thermal units is the quantity of heat exchanged between the metal and the steam during expansion.

R_e thermal units is the quantity of heat exchanged between the metal and the steam during exhaust.

R_c thermal units is the quantity of heat exchanged between the metal and the steam during compression.

R_a, R_d, R_c will have positive sign when the heat passes from the steam to the metal. R_e, on the other hand, is positive when the heat passes from the metal to the steam.

In general, R_c and R_a are positive, and R_d is negative; that is to say, that, generally, the steam warms the metal during the compression and admission, and the metal gives up its heat to the steam during the expansion; but, with a gas flame jacket, it has already been found that the metal received from outside sufficient quantity of heat to superheat the steam even during admission. Whatever it may be, the exchange which takes place while the cylinder has no communication with the condenser will be called R_f. It follows from this definition that—

$$R_f = R_c + R_a + R_d, \quad . \quad . \quad . \quad . \quad . \quad (11.)$$

The entire exchange for one stroke of the piston can be called R without exponent, and, therefore,—

$$R = R_f - R_e = R_c + R_a + R_d - R_e, \quad . \quad . \quad . \quad (12.)$$

This total exchange would be zero if there were no heat to the amount denoted by E, which passes through the metal, and is lost by radiation outside in the cases when the machines were jacketed. For these latter, necessarily, $R = E$. When the jacket furnishes Q' thermal units as a total, there will go E outwards and $-R$ inwards; whence, $Q' = E + (-R)$.

The second fundamental equation of the practical theory can then be written as follows:

$$R = E - Q', \text{ or } R_f + Q' = E + R_e;$$

or, again,—

$$R_c + R_a + R_d = R_e + (E - Q'), \quad . \quad . \quad . \quad \text{(II.)}$$

23. The quantities which are designated by these initial letters R are not given directly by experiment; they must be calculated, which requires four new equations, in which R_c, R_a, R_d, R_e will be the unknown quantities. For the equations concerning the expansion and the compression they are easy to write, since the weight of fluid in action is constant. The fluid is enclosed

in the cylinder, and cannot receive thermal units, except from the metal, or give them out, except to the metal. As we have seen in equation No. 7, where U_0 and U_1 represent the internal heat of the steam at the commencement and at the end of expansion, we shall have

$$U_0 - U_1 = T_d + R_d.$$

Similarly, the internal heat of the fluid at the commencement of compression was U_2. The heat T_c resulting from the work of compression is added to this, and this sum ought to preserve for the steam the heat denoted by U_3, and also to give R_c thermal units to the metal; whence,

$$U_2 + T_c = U_3 + R_c.$$

In the matter of the periods of admission and exhaust, the problem is complicated by the fact that the steam, in coming into the cylinder, carries thither thermal units to the amount Q, and, in leaving the cylinder, it carries out $(C + c)$ thermal units, to drop them into the condenser, or dissipate them in the outer air. From whence we draw the conclusion that for the admission

$$U_3 + Q = U_0 + T_a + R_a;$$

and for the exhaust,

$$U_1 + R_e + T_e = U_2 + (C + c).$$

These last four equations can be written as follows, and adding to them the preceding ones, we have

$$Q + Q' = T + (C + c) + E \quad . \quad . \quad . \quad . \quad \text{(I.)}$$
$$R_c + R_a + R_d - R_e = R = E - Q' \quad . \quad . \quad \text{(II.)}$$
$$R_a = U_3 + Q - U_0 - T_a \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(III.)}$$
$$R_d = U_0 - U_1 - T_d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(IV.)}$$
$$R_e = U_2 + (C + c) - U_1 - T_e \quad . \quad . \quad . \quad . \quad \text{(V.)}$$
$$R_c = U_2 - U_3 + T_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(VI.)}$$

24. In connection with R_a, the remark should be made that its quantity does not represent solely the exchange of heat between the vapor and the metal. There is heat given out by the steam to compress that which under low pressure filled the waste spaces at the end of the exhaust. This heat, which is not shown by the indicator, is, according to these equations, an integrant

part of that which we have called R_a. In reference to U_2 it should be remembered that it has been obtained by the hypothesis that at the commencement of compression there is only in the cylinder steam which is dry and saturated.

25. The object of the calculations in our theory is to obtain the values of R_c, R_a, R_d, R_e, R_f, and to represent these values graphically. For this purpose the same scale is adopted for the exchange of heat as for the diagrams of pressure, and in the following manner: T_a represents a certain number of thermal units lost by the steam while the piston is generating the volume V_a cubic feet. In like manner R_a represents the thermal units lost by the steam while the piston sweeps through the same volume. The value of T_a is represented on the diagram by a surface whose length, representing V_a, is the base. If the pressure during admission was constant and equal to p_a, this diagram would be a horizontal line at the height which is represented by p_a, and the area would be rectangular and equal to $p_a V_a$. If p_a is counted in pounds per square foot, then $p_a V_a = 772 T_a$; whence $p_a = \dfrac{772 T_a}{V_a}$.

Similarly a height r_a can be calculated such that

$$r_a V_a = 772 R_a,$$

whence

$$r_a = \frac{772 R_a}{V_a};$$

also, in like manner,

$$r_d = \frac{772 R_d}{V_d},$$

$$r_e = \frac{772 R_e}{V_e},$$

$$r_c = \frac{772 R_c}{V_c}.$$

26. If the exchanges are positive, that is to say, if it is the steam which furnishes heat to the metal, the ordinates r will be carried above the axis in the forward stroke, and below in the backward stroke. If the exchanges are negative, that is to say, if it is the metal which furnishes heat to the steam, the ordinates r are carried below the axis for forward stroke, and above it for the backward. In a trial, of which mention will be made hereafter, R_a has been found positive; R_d negative; R_e negative; and R_c

positive. Then the diagram of exchanges will have the form shown in Figure 4:

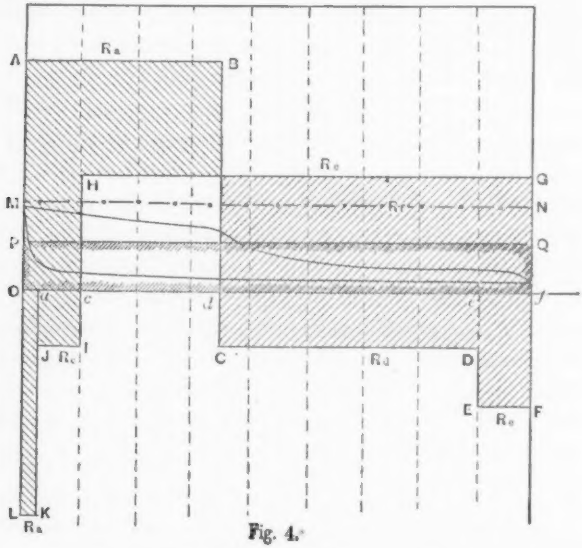


Fig. 4.

The area		$aKLABda$	represents	R_a,
"	"	$dCDed$	"	R_d,
"	"	$eEFGHce$	"	R_e,
"	"	$cIJac$	"	R_c.

27. The positive exchange is represented by the surface covered with hatchings from left to right. The negative transfer is represented by the surface with hatchings through right to left. The difference of these two surfaces thus distinguished would be zero if there were no heat lost by external radiation or received from a jacket. In the example hereabove there was a loss. On the preceding diagram we have added a straight line MN, drawn in alternate dots and lines at a height such that the surface $OMNfO$ represents $R_f = R_c + R_a + R_d$. This is the loss due to the action of the walls. The straight line PQ is at such a height that, at the same scale, the surface $OPQfO$, of which the contour is edged by hatchings, represents the positive work T_f of the pressure of the steam.

In order to institute a comparison of all engines among each other of whatsoever dimensions or mode of action, we refer all the quantities to one pound of steam employed. We will show the method by treating an example.

APPLICATION I.

28. The dimensions taken directly from the machine give the following volumes expressed in cubic feet :

$v = 0.1766$	and therefore :
$V_0 = 2.8183$	$V_a = 2.6417$
$V_1 = 17.3051$	$V_d = 14.4868$
$V_2 = 1.4127$	$V_e = 15.8924$
$V_3 = 0.1766$	$V_c = 1.2361$
$V_4 = 17.3051$	$V_f = 17.1285$

Then the measures taken on the mean diagram give the pressures in pounds per square foot and the work in thermal units:

Pounds per square foot.	Thermal Units.
$P_0 = 7858$	$T_a'' = 0$
$P_1 = 1175$	$T_a' = 29.76$
$P_2 = 340$	$T_a = 29.76$
$P_3 = 1270$	$T_d = 54.37$
	$T_e' = 0$
	$T_e'' = 8.73$
	$T_e = 8.73$
	$T_c = 0.91$
	$T_f = 84.13$
	$T_n = 9.64$
	$T = 74.49$

The other data from experiment are the weight of wet steam delivered from the boiler, $M_a = 0.5807$ lbs.

The weight of pure steam which it contains, $m = 0.5741$ "

And the weight of water in it, $M - m = 0.0066$ "

The mean pressure in the boiler, $P = 10180$ " sq.ft.

The tables give for this pressure, the total heat of the steam per lb., $\lambda = 1174.47$ t. u.

The heat of the water per lb., $q = 274.05$ " "

The heat carried to the cylinder is calculated by the formula

$$Q = m\lambda + (M_a - m)\,q. \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which gives

$$Q = 674.26 + 1.81. \qquad\qquad Q = 676.07.$$

Other data.

Weight of cold water of condensation per stroke,	$M_e = 19.6502$ lbs.
Initial heat of this water per lb.,	$q_i = 29.70$ t. u.
Final heat of this water after condensation per lb.,	$q_f = 58.05$ "
By calculation $C = M_e\,(q_f - q_i)$,	$C = 557.08$, "
$c = M_a\,q_f$,	$c = 33.71$, "
	$C + c = 590.79$ t. u.
The exterior radiation per stroke,	$E = 7.94$ "

29. To verify the trial by equation (I).

$$T = 74.49 \qquad\qquad Q = 676.07$$
$$(C + c) = 590.79 \qquad\qquad 673.22$$
$$E = 7.94 \qquad\qquad \text{The error } 2.85$$
$$\text{Total } 673.22 \qquad\qquad \text{which is } 0.42 \text{ per cent.,}$$

a very close approximation.

To calculate the weight of steam which remains in the clearance when the exhaust is just closed, the data will be

$V_2 = 1.4127$ cubic feet, and $P_2 = 340$ lbs. per square foot.

From the tables the weight of a cubic foot is deduced:

$$\delta_2 = 0.00675 \text{ lbs.}$$

whence

$$m_2 = V_2\,\delta_2 = M_c = 0.0095 \text{ lbs.}$$

Whence the weight of fluid in action during expansion

$$(M_a + M_c) = 0.5902.$$

The weight of fluid in action during compression

$$M_c = 0.0095 \text{ lbs.}$$

30. The following calculations are made by means of the tables in order to determine the values of internal heat of the fluid, which we have called respectively U_0, U_1, U_2, U_3.

	U_0.		U_1.	
Data	V_0	$= 2.8183$	V_1	$= 17.3051$
Data	P_0	$= 7858$	P_1	$= 1175$
Given by the tables	δ_0	$= 0.13009$	δ_1	$= 0.02180$
Given by the tables	$_0$	$= 256.64$	q_1	$= 152.35$
Given by the tables	ρ_0	$= 834.58$	ρ_1	$= 915\ 85$
Calculated	m_0	$= 0.3666$	m_1	$= 0.3773$
Calculated	$m_0\rho_0$	$= 305.96$	$m_1\rho_1$	$= 345.55$
Calculated	$(M_a + M_c)\,q_1$	$= 151.47$	$(M_a + M_c)\,q_1$	$= 89.92$
Calculated	U_0	$= 457.43$	U_1	$= 435\ 47$

	U_2.			U_3.		
Data	V_2	=	1.4127	V_3	=	0.1766
Data	P_2	=	340	P_3	=	1270
Given by the tables	δ_2	=	0 00675	δ_3	=	0.02345
Given by the tables	q_2	=	100.48	q_3	=	155.95
Given by the tables	ρ_2	=	956 63	ρ_3	=	913.02
Calculated	m_2	=	0.0095	m_3	=	0.0041
Calculated	$m_2\rho_2$	=	9.09	$m_3\rho_3$	=	3.74
Calculated	$M^c q_2$	=	0.95	$M^c q_3$	=	1.48
Calculated	U_2	=	10.04	U_3	=	5.22

31. The calculation of the exchanges denoted by R are made according to the formulas (III.), (IV.), (V.), and (VI.), by means of the preceding data.

Thermal Units.

$R_a = 5.22 + 676.07 - 457.43 - 29.76.$ $\qquad R_a = 194.10.$
$R_d = 457.43 - 435.47 - 54.37.$ $\qquad R_d = -32.41.$
$R_e = 10.04 + 590.79 - 435.47 - 8.73.$ $\qquad R_e = 156.63.$
$R_c = 10.04 - 5.22 + 0.91.$ $\qquad R_c = 5.73.$

Whence,

$R_f = R_c + R_a + R_d = 5.73 + 194.10 - 32.41.$ $\qquad R_f = 167.42.$
$R_e + E = 156.63 + 7.94 = 164.57 = R_f - 2.85.$

32. It is apparent that the percentage of steam in the mixture, which was $x = \frac{0.5741}{0.5807} = 0.989$ before entering the cylinder, has fallen to $x_0 = \frac{0.3666}{0.5902} = 0.621$ at the end of admission. During the expansion this value rises to $x_1 = \frac{0.3773}{0.5902} = 0.639$. There was, therefore, an evaporation of 0.0107 lbs.

At the beginning of the compression the steam is supposed to be dry $x_2 = 1$ and $m_2 = 0.0095$. At the end $m_3 = 0.0041$ and $x_3 = \frac{0.0041}{0.0095} = 0.432$. Hence the effect of compression has been a condensation of 0.0054 lbs.

33. With a view of comparing the diagrams of steam-pressure and pressure equivalent to the heat exchange for all engines, all heat quantities are referred to a well defined unit, namely, 1179.12 thermal units, the total heat of 1 lb. of steam at 6 atmospheres, or 88.2 lbs. per sq. inch, which is about an average of the pressures used. The total heat used per stroke, divided by the total heat of one pound of steam, is termed *the weight of steam really expended*. We denote it by π lbs.

Then $$\pi = \frac{Q + Q'}{1179.12}, \text{ or } \frac{Q}{1179.12}$$

according as the jacket has furnished Q' thermal units or has been out of action.

Now to trace the diagrams let a constant base be taken two inches in length (or 50 millimetres in metric units) which represents the volume which has been called V_f, that is to say, the volume swept through by the piston in one stroke. The quantities of heat $\frac{T}{\pi}$ or $\frac{R}{\pi}$, considered during the time that the piston is generating this volume V can be expressed in lbs. per sq. ft. per lb. by the expressions

$$\frac{772\ T}{\pi.\ V} \text{ and } \frac{772\ R}{\pi.\ V}$$

But to bring them to the same units for all engines, the division must be made not by V, but by the ratio $\frac{V}{V_f}$, which is the same thing as multiplying the preceding expressions by V_f. The ordinates of the diagrams will be then proportional to the values of the expressions

$$\frac{772\ T}{\pi \cdot V} V_f \quad \text{and} \quad \frac{772\ R}{\pi\ V} V_f.$$

Nothing prevents the representation of these quantities at an agreed scale. A suitable one is one millimetre for 10,000 units. In the example which has been treated there will result successively,

$$Q = 676.07 \qquad \pi = \frac{676.07}{1179.12} = 0.5734.$$

$$\frac{772\ T_a}{\pi\ V_a} V_f = 259{,}794 \qquad \frac{772\ R_a}{\pi\ V_a} V_f = 1{,}694{,}420.$$

$$\frac{772\ T_d}{\pi\ V_d} V_f = 86{,}550 \qquad \frac{772\ R_d}{\pi\ V_d} V_f = -\ 51{,}592.$$

$$\frac{772\ T_e}{\pi\ V_e} V_f = 12{,}668 \qquad \frac{772\ R_e}{\pi\ V_e} V_f = 227{,}282.$$

$$\frac{772\ T_c}{\pi\ V_c} V_f = 16{,}977 \qquad \frac{72\ R}{\pi\ V_c} V_f = 106{,}901.$$

$$\frac{772\ T_f}{\pi\ V_f} V_f = 112{,}113 \qquad \frac{772\ R_f}{\pi\ V_f} V_f = 223{,}106.$$

34. The diagram, Fig. 1, hereafter, has been drawn from these data.

The height of the rectangle whose area represents R_a is 169.4 millimetres.

The height of the rectangle whose area represents R_d is 5.2 millimetres.

The height of the rectangle whose area represents R_e is 22.7 millimetres.

The height of the rectangle whose area represents R_c is 10.7 millimetres.

The height of the rectangle whose area represents R_f is 22.3 millimetres.

For the diagram representing the work T_a, T_d, etc., the form given by the indicator itself is retained as much as possible, and indeed attention is called to the following method of getting the ordinates which correspond to a given pressure. In the expressi $\frac{772\,T}{\pi\,V} V_f$, the factor $\frac{772\,T}{V}$ represents lbs. per sq. ft. Hence it is apparent that it is enough to multiply the pressure expressed in lbs. per sq. ft. by $\frac{V_f}{\pi}$, or to divide by $\frac{\pi}{V_f}$, to obtain the height of the ordinate which will represent it at the desired scale for the diagram. For example, the pressure at the commencement of expansion was 7,852 lbs. per sq. ft., and at the end of expansion it was 1,175 lbs. per sq. ft. The ratio $\frac{V_f}{\pi} = \frac{17.1285}{0.5734}$ $= 29.872$. Hence, to represent the pressure at the beginning we should have 234,555 lbs. per sq. ft., and at the end, 35,100 lbs. per sq. ft., which would be in millimetres in height, 23.5 and 3.5. The mean heights representing the works are as follows:

T_a,	26.0	millimetres.
T_d,	8.7	"
T_e,	1.3	"
T_c,	1.7	"
T_f,	11.2	"

35. *Note.*—The preceding figures may be translated immediately in metric measures by multiplying them by the co-efficient 0.305, so that the quantities in kilogrammes per square metre

per kilogramme after multiplication by V_f cubic metres become as follows :

R_a, 516798	T_a, 79237	P_0, 234734
R_d, 15736	T_d, 26398	P_1, 35100
R_e, 69321	T_e, 3864	P_2, 10156
R_c, 32605	T_c, 5178	P_3, 37937
R_f, 68047	T_f, 34194	

Hence it appears that if a study of a trial has been made in metric measures, to obtain a diagram which can be superposed on that which would have resulted had the calculations been made in English measures, it will be enough to represent 3050 units in metric measures by one millimetre of height.

Application II.

36. The dimensions taken directly from the machine are in cubic feet.

$v = 0.0457$

$V_0 = 0.2879$	$V_a = 0.2422$
$V_1 = 0.4197$	$V_d = 0.1318$
$V_2 = 0.1298$	$V_e = 0.4373$
$V_3 = 0.0517$	$V_c = 0.0781$
$V_4 = 0.4934$	$V_f = 0.4477$

The measurements on the indicator diagram have given :

$P_0 = 3370$ lbs. per sq. f.	$T''_a = 0.0187$ th. un.	$T'_e = 0.1833$ th. un.
$P_1 = 2448$ "	$T'_a = 1.2290$ "	$T''_e = 0.4071$ "
$P_2 = 886$ "	$T_a = 1.2103$ "	$T_e = 0.2238$ "
$P_3 = 2246$ "	$T_d = 0.4683$ "	$T_c = 0.1413$ "

whence $T_f = 1.8806$, $T_n = 0.5671$, $T = 1.3135$.

Experience would add the following data : the weight of steam used at each piston stroke, $M_a = 0.020$ lbs. This steam is dry and at the pressure of the boiler, 8064 lbs. per sq. ft. The external radiation is $E = 0.5$ thermal units.

37. The water of condensation was not measured, nor its temperature before and after condensation. Hence the verification pointed out by equation (I.) cannot be made, since the quantity $(C + c)$ is not known. Similarly the equation (V.) cannot give the value of R_e.

In this case we use another method, necessary, moreover, when

there is no condensation. This method is based on the hypothesis that the heat furnished by the walls during exhaust, or R_e, is entirely used in evaporating the water mixed with the steam at the moment when exhaust began. If, then, M_a is the weight of steam coming from the boiler, M_c the weight of steam remaining in the cylinder when the exhaust ceases, and m_1 the weight of pure steam contained in the mixture at the moment where exhaust begins, the weight of hot water present in the mixture at this moment will be

$$(M_a + M_c - m_1).$$

The internal heat necessary to vaporize this water at the pressure P_1 is what we have called ρ_1, and calling R_1 the value of R_e, calculated according to the hypothesis of the author, the value results:

$$R_1 = (M_a + M_c - m_1)\,\rho_1 \quad . \quad . \quad . \quad . \quad . \quad \text{(VII.)}$$

The value of $(C + c)$ is deducted from the equation (V) by replacing U_1 by its value $(M_a + M_c)\,q_1 + m_1\rho_1$. This value will be designated by $(C_1 + c)$.

$$(C_1 + c) = R_1 + U_1 - U_2 + T_e,$$
$$(C_1 + c) = (M_a + M_c - m_1)\,\rho_1 + (M_a + M_c)\,q_1 + m_1\rho_1 - M_c\,(q_2 + \rho_2) + T_e$$
$$(C_1 + c) = M_a\,(q_1 + \rho_1) + M_c\,[(q_1 + \rho_1) - (q_2 + \rho_2)] + T_e. \quad \text{(VIII.)}$$

The value of $(C_1 + c)$ can be determined and introduced in the equation I. when experiment has given what is necessary to calculate the quantities which are found in the second member of the equation VIII. Here are the calculations:

$$M_c = m_2 = V_2\delta_2\text{; and } P_2 = 886.$$

The tables give

$$\delta_2 = 0.01670\text{; moreover } V_2 = 0.1298,$$

then

$$M_c = 0.00217\text{;}$$

and moreover,

$$q_2 = 139.63\text{; } \rho_2 = 925.85\text{;}$$

whence

$$q_2 + \rho_2 = 1065.48.$$

Because

$$P_1 = 2448,$$

it is found in the tables,

$$q_1 = 188.41\,;\ \rho_1 = 887.63\,;$$

whence

$$q_1 + \rho_1 = 1076.04.$$

Whence we find successively

$$M_a(q_1 + \rho_1) = 21.5208,$$
$$M_c[(q_1 + \rho_1) - (q_2 + \rho_2)] = 0.0229.$$

From experience,

$$T_e = 0.2238.$$

Whence (VIII.):

$$(C_1 + c) = 21.5208 + 0.0229 + 0.2238 = 21.7675.$$

Now, to make the verification indicated by the equation (I), Q must be calculated, or since the steam is dry when it comes from the boiler, $Q = M_a\lambda$. The pressure being 8.064 lbs. per sq. ft., the tables give $\lambda = 1169.80$ and consequently

$$Q = M_a\lambda = 0.02 \times 1169.80 = 23{,}3960.$$

Among the data we have

$$T = 1.3135,$$
$$E = 0.5000\,;$$

Adding to it

$$(C_1 + c) = 21.5208,$$

we shall have

$$(C_1 + c) + T + E = 23{,}3343.$$

The difference between this and Q is 0.0623, or about $\frac{1}{4}$ per cent.

From what has immediately preceded we can deduce U_1 and U_2,

$$U_1 = (M_a + M_c)\,q_1 + m_1\rho_1 = 0.02217 \times 188{,}41 + m_1 \times 887.63.$$

and $$m_1 = V_1\delta_1 = 0{,}4197 \times 0.04352 = 0.01827,$$

whence $$U_1 = 4.1770 + 16{,}2170 = 20.3940.$$

But $$U_2 = M_c(q_2 + \rho_2) = 2{,}3121,$$

Let us now compute U_0 and U_3. Since $P_0 = 3370$, the tables give

$$\delta_0 = 0.05877,\quad q_o = 205.62,\quad \rho_0 = 874.20.$$

whence $$m_0 = V_0\delta_0 = 0.01692,$$

$$U_0 = (M_a + M_e)\,q_0 + m_0\rho_0 = 4.5586 + 14{,}7915 = 19.3501,$$

The tables give for $P_3 = 2246$:

$$\delta_3 = 0.04013,\ q_3 = 183{,}92,\ \rho_3 = 891{,}12;\ \text{and}\ V_3 = 0.0517$$

whence $m_3 = V_3\,\delta_3 = 0.00207$; it follows that the steam is wet since it contains 0.00207 of steam and 0.00010 of water. Hence we have

$$U_3 = M_c\,q_3 + m_3\,\rho_3 = 0.3991 + 1.8446 = 2.2437.$$

Finally, computing the values of the exchanges of heat R,

$$\begin{aligned} R_a &= U_3 + Q - U_0 - T_a = 5.0793, \\ R_d &= U_0 - U_1 - T_d = -1.5122, \\ R_e &= U_2 + (C + c) - U_1 - T_e = 3.2151, \\ R_c &= U_2 - U_3 + T_c = 0.2097, \end{aligned}$$

whence $R_f = 3.7768.$

Tracing now the diagram of exchanges, we will first compute π;

$$\pi = \frac{23{,}3960}{1179.12} = 0.019842,$$

whence we derive the products

$$\frac{772}{\pi}\frac{T}{V}\,V_f \text{ and } \frac{772}{\pi}\frac{R}{V}\,V_f,$$

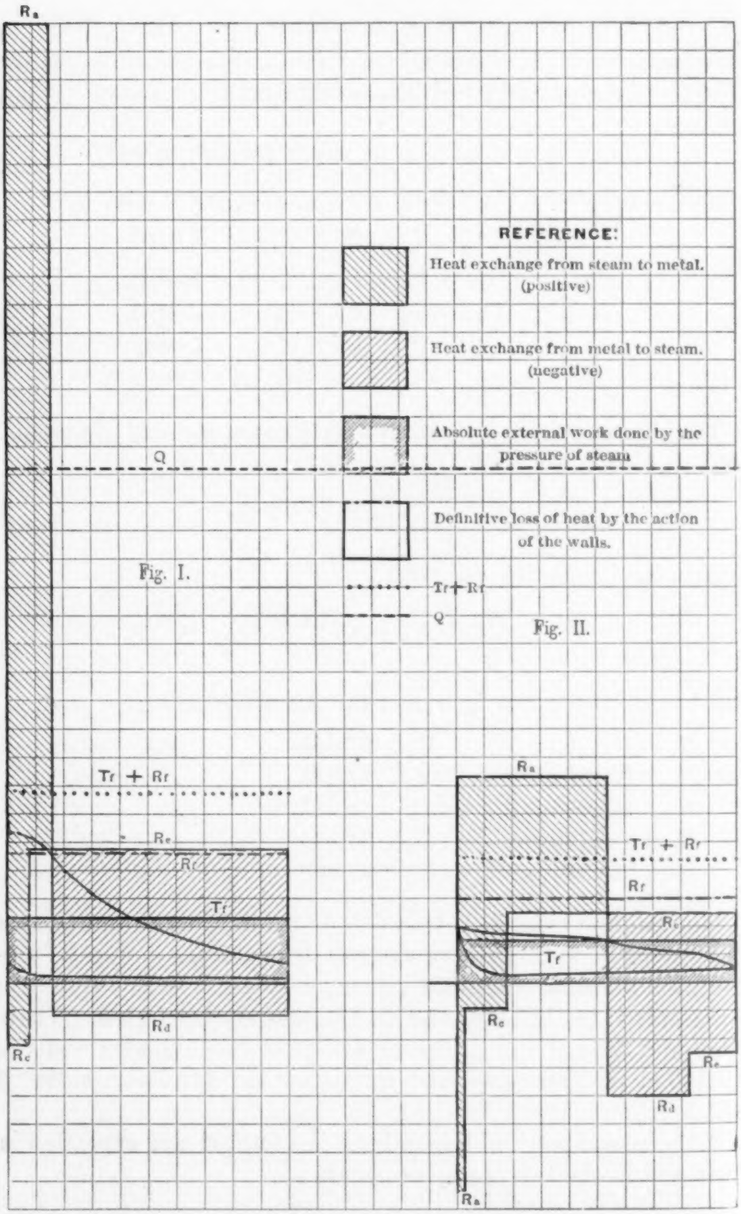
which give finally the figures in the following table:

R_a	365,299	T_a	87,044	P_0	76,038
R_d	199,854	T_d	61,891	P_1	55,235
R_e	128,066	T_e	8,914	P_2	19,991
R_c	46,770	T_c	31,514	P_3	·50,677
R_f	146,945	T_f	73,169		

The diagram (Fig. 2) is traced from these figures. It appears that it differs in a striking manner from diagram Fig. 1. Indeed, the absolute work per pound of steam is in the first case represented by 112,113, and in the second by 73,169. Their ratio is 1.53, and the injurious action of the cylinder walls is 223,106 in the first case, and in the second 146,945, a ratio of 1.52.

If the actual work obtained from the total steam expended is compared in the two cases, we obtain

In the first case, $\frac{T_f}{Q} \parallel 0.124,$


REFERENCE:
Heat exchange from steam to metal. (positive)
Heat exchange from metal to steam. (negative)
Absolute external work done by the pressure of steam
Definitive loss of heat by the action of the walls.
Tr + Rr
Q
Fig. I.
Fig. II.
Ra
Tr + Rr
Rc
Rd
Re
Rf
Tr
Q

and in the second, $\frac{T_f}{Q} = 0.080$;

afterwards, first case, $\frac{R_f}{Q} = 0.248$,

and second case, $\frac{R_f}{Q} = 0.161$.

It should be noted that also

in the first case $\frac{U_1}{Q} = 0.644$,

and in the second case $\frac{U_1}{Q} = 0.872$.

To take account of the phenomenon which occurs in the engine it is convenient to make the following table :

	I.	II.
Absolute work in fraction of Q	0.124	0.080
Loss by wall in fraction of Q	0.248	0.161
Other losses in fraction of Q	0.628	0.759
U_1 = heat of the steam at beginning of exhaust in fraction of Q	0.644	0.872

The quantity of heat Q can be represented in the same way as the others by a rectangular surface whose base is that of the diagram, and whose height will be represented by

$$\frac{772}{\pi}\frac{Q}{V_f}V_f = \frac{772\,Q}{\pi},$$

But

$$\pi = \frac{Q}{1179.12}.$$

Whence the height is $\frac{772Q}{Q} \times 1179.12 = 910.281$.

On the diagram this quantity is represented by a height about 91 millimetres.

In the first figure annexed we have traced a straight horizontal line broken at this height. Another dotted line has been added at the height denoted by $T_f + R_f$. The distance between these two latter lines gives an idea of the losses other than those which are due to the influence of the cylinder walls. The diagram translates to the eye the table just given above, in order to make the comparison of the two trials here concerned.

It appears at first from this table that the great loss comes from the fact that the steam is surrendered when it still contains a considerable quantity of heat, U_1, and the sum of the losses other than by wall's action come very nearly equal to U_1. It is because the period of expansion being short in the second example, that the steam did not have time to do much work at the expense of its heat. More prolonged expansion in the first case, has diminished the value of U_1 by about 20 per cent. below its value in the second case; thus the sum of the losses is thirteen per cent. greater in the second case than in the first; but of this 13 per cent. 4 only go to the work and 9 to the loss by the walls. It should be remarked also that the influence of the clearance enters to some extent, but it was taken in the value of the losses through the walls by reason of ignorance how to isolate it. It would appear possible to affirm that in neither of these two cases did the piston have any leakage, but it should be remarked that the leakages would not be able to be distinguished in the calculation from the action of the walls. Indeed, it is immaterial whether the heat passes from one side to the other side of the piston conveyed by the steam or by the metal.

These examples will, doubtless, be sufficient to show the working of the tables of steam, and are respectfully submitted to the American Society of Mechanical Engineers.

1	2	3	4	5	6	7	8	9	10
Pressure in lbs. per sq. in.	Pressure in lbs. per sq. ft.	Temperature in degrees F.	Volume of one lb. of pure saturated steam, in cubic feet.	Weight of one cubic foot of pure saturated steam, in lbs.	Heat of one lb. of saturated water, in thermal units.	Internal latent heat of one lb. of pure saturated steam, in thermal units.	External latent heat of one lb. of pure saturated steam, in thermal units.	Total latent heat of one lb. of pure saturated steam, in thermal units. $r = \rho + APu$.	Total heat of one lb. of saturated steam, in lbs. thermal units. $\lambda = q + r$. $\lambda = q + \rho + APu$.
p	P	t	v	δ	q	ρ	APu	r	λ
1.00	144	102.00	334.23	0.00299	70.00	980.62	62.34	1042.96	1113.05
1.25	180	109.56	270.47	0.00370	77.67	974.63	63.06	1037.69	1115.36
1.50	216	115.90	227.51	0.00440	84.03	969.61	63.65	1033.26	1117.29
1.75	252	121.40	196.60	0.00509	89.55	965.25	64.17	1029.42	1118.97
2.00	288	126.27	173.23	0.00577	94.44	961.39	64.62	1026.01	1120.45
2.50	360	134.60	140.22	0.00713	102.81	954.80	65.38	1020.18	1122.99
3.00	432	141.62	117.98	0.00848	109.88	949.24	66.01	1015.25	1125.13
3.50	504	147.70	102.00	0.00980	115.99	944.42	66.58	1011.00	1126.99
4.00	576	153.07	89.80	0.01112	121.40	940.17	67.06	1007.23	1128.63
1	144	102.00	334.23	0.00299	70.09	980.62	62.34	1042.96	1113.05
2	288	126.27	173.23	0.00577	94.44	961.39	64.62	1026.01	1120.45
3	432	141.62	117.98	0.00848	109.88	949.24	66.58	1011.00	1126.99
4	576	153.07	89.80	0.01112	121.40	940.17	67.06	1007.23	1128.63
5	720	162.33	72.50	0.01373	130.72	932.84	67.89	1000.73	1131.45
6	864	170.12	61.10	0.01631	138.58	926.67	68.58	995.25	1133.83
7	1008	176.91	53.00	0.01887	145.43	921.29	69.18	990.47	1135.90
8	1152	182.91	46.60	0.02140	151.48	916.54	69.71	986.25	1137.73
9	1296	188.32	41.82	0.02391	156.94	912.25	70.18	982.43	1139.38
10	1,440	193.24	37.80 *	0.02641	161.92	908.35	70.61	978.96	1140.88
11	1,584	107.77	34.61	0.02889	166.50	904.78	70.99	975.76	1142.26
12	1,728	201.96	31.90	0.03136	170.74	901.47	71.34	972.80	1143.54
13	1,872	205.86	29.58	0.03381	174.68	898.37	71.68	970.05	1144.73
14	2,016	209.55	27.59	0.03625	178.43	895.43	72.00	967.43	1145.86
15	2,160	213.03	25.87	0.03868	181.94	892.67	72.29	964.97	1146.91
16	2,304	216.30	24.33	0.04110	185.25	890.08	72.57	962.66	1147.91
17	2,448	219.41	22.98	0.04352	188.41	887.63	72.82	960.45	1148.86
18	2,592	222.38	21.78	0.04592	191.41	885.29	73.07	958.35	1149.76
19	2,736	225.20	20.70	0.04831	194.29	883.05	73.30	956.34	1150.63
20	2,880	227.92	19.7200	0.05070	197.03	880.90	73.53	954.42	1151.45
21	3,024	230.52	18.8412	0.05308	199.68	878.83	73.74	952.57	1152.25
22	3,168	233.02	18.0342	0.05545	202.22	876.85	73.94	950.79	1153.01
23	3,312	235.43	17.2951	0.05782	204.67	874.94	74.13	949.07	1153.74
24	3,456	237.75	16.6176	0 06018	207.03	873.10	74.32	947.42	1154.45
25	3,600	240.00	15 9923	0.06253	209.31	871.32	74.51	945.83	1155.14
26	3,744	242.18	15.4154	0.06487	211.52	869.60	74.69	944.28	1155.80
27	3,888	244 28	14.8782	0.06721	213.67	867.93	74.85	942.78	1156.45
28	4,032	246.33	14.3780	0.06955	215.75	866.31	75.01	941.32	1157.07
29	4,176	248.31	13.9124	0.06188	217.76	864.74	75.17	939.91	1157.67
30	4,320	250.25	13.4771	0.07420	219.73	863.22	75.33	938.53	1158.26
31	4,464	252.12	13.0677	0.07652	221.65	861.72	75.47	937.19	1158.84
32	4,608	253.95	12.6839	0.07884	223.51	860.27	75.61	935.88	1159.39
33	4,752	255.74	12.3229	0.08115	225.33	858.86	75.76	934.61	1159.94
34	4,896	257.48	11.9823	0.08346	227.10	857.48	75.89	933.37	1160.47
35	5,040	259.18	11.6603	0.08576	228.84	856.13	76.02	932.15	1160.99
36	5,184	260.84	11.3579	0.08806	230.52	854.81	76.16	930.97	1161.49
37	5,328	262.46	11.0686	0.09035	232.18	853.53	76.28	929.81	1161.99
38	5,472	264.05	10.7947	0.09264	233.80	852.27	76.40	928.67	1102.47
39	5,616	265.60	10.5341	0.09493	235.39	851.04	76.52	927.56	1162.95
40	5,760	267.12	10.2870	0.09721	236.94	849.84	76.63	926.47	1163.41
41	5.904	268.61	10.0517	0.09949	238.46	848.66	76.75	925.41	1163.87
42	6,048	270 07	9.8261	0.10177	239.95	847.50	76.86	924.36	1164.31
43	6,192	271.51	9.6117	0.10404	241.42	846.36	76.97	923.33	1164.75
44	6,336	272.92	9.4065	0.10631	242.86	845.25	77.07	922.32	1165.18
45	6,480	274.30	9.2109	0.10857	244.27	844.15	77.18	921.33	1165.60
46	6,624	275.65	9.0228	0.11083	245.65	843.07	77.29	920.36	1166.01

1	2	3	4	5	6	7	8	9	10
Pressure in lbs. per sq. in.	Pressure in lbs. per sq. ft.	Temperature in degrees F.	Volume of one lb. of pure saturated steam, in cubic feet.	Weight of one cubic foot of pure saturated steam, in lbs.	Heat of one lb. of saturated water, in thermal units.	Internal latent heat of one lb. of pure saturated steam, in thermal units.	External latent heat of one lb. of pure saturated steam, in thermal units.	Total latent heat of one lb. of pure saturated steam, in thermal units. $r = \rho + APu$.	Total heat of one lb. of saturated steam, in thermal units. $\lambda = q + r$, $\lambda = q + \rho + APu$.
p	P	t	v	δ	q	ρ	APu	r	λ
47	6,768	276.99	8.8425	0.11309	247.01	842.02	77.39	919.41	1166.42
48	6,912	278.30	8.6693	0.11535	248.35	840.98	77.49	918.47	1166.82
49	7,056	279.59	8.5034	0.11760	249.67	839.96	77.58	917.54	1167.21
50	7,200	280.85	8.3438	0.11985	250.97	838.96	77.67	916.63	1167.60
51	7,344	282.10	8.1901	0.12210	252.24	837.98	77.76	915.74	1167.98
52	7,488	283.33	8.0422	0.12434	253.49	837.01	77.85	914.86	1168.35
53	7,632	284.53	7.8999	0.12658	254.73	836.05	77.94	913.99	1168.72
54	7,776	285.72	7.7628	0.12882	255.96	835.10	78.03	913.13	1169.09
55	7,920	286.90	7.6207	0.13105	257 15	834.18	78.12	912.29	1169.44
56	8,064	288.06	7.5030	0.13328	258.34	833.25	78.21	911.46	1169.80
57	8,208	289.20	7.3795	0.13551	259.50	832.35	78.29	910.64	1170.14
58	8,352	290.32	7.2600	0 13774	260.66	831.46	78.37	909.83	1170.49
59	8,496	291.43	7.1445	0.13997	261.79	830.58	78.45	909.03	1170.82
60	8,640	292.52	7.0328	0.14219	262.91	829.72	78.53	908.25	1171.16
61	8,784	293.60	6.9248	0.14441	264.02	828.86	78 61	907.47	1171.49
62	8,928	294.66	6.8199	0.14663	265.11	828.02	78.68	906.70	1171.81
63	9,072	295.71	6.7182	0.14885	266.18	827.19	78.76	905.95	1172.13
64	9,216	296.75	6.6194	0.15107	267.25	826.37	78.83	905.20	1172.45
65	9,360	297.78	6.5236	0.15329	268.30	825.56	78.90	904.46	1172.76
66	9,504	298.79	6.4305	0.15551	269.34	824.76	78.97	903.73	1173.07
67	9,648	299.79	6.3405	0.15772	270.37	823.97	79.04	903.01	1173.38
68	9,792	300.78	6.2530	0.15992	271.38	823.19	79.11	902.30	1173.68
69	9,936	301.75	6.1681	0.16212	272.38	822.41	79.18	901.59	1173.97
70	10,080	302.72	6.0857	0.16432	273.37	821.65	79.25	900.90	1174.27
71	10,224	303.67	6.0053	0.16652	274.35	820.89	79.32	900.21	1174.56
72	10,368	304.62	5.9274	0.16871	275.32	820.14	79.39	899.53	1174.85
73	10,512	305.55	5.8515	0.17090	276.28	819.40	79.46	898.85	1175.13
74	10,656	306.47	5.7773	0.17309	277.22	818.66	79.53	898.19	1175.41
75	10,800	307.39	5.7052	0.17528	278.16	817.94	79.59	897.53	1175.69
76	10,944	308.29	5.6346	0.17747	279.09	817.23	79.65	896.88	1175.97
77	11,088	309.18	5.5661	0.17966	280.01	816.52	79.71	896.23	1176.24
78	11,232	310.07	5.4990	0.18185	280.92	815.82	79.77	895.59	1176.51
79	11,376	310.95	5.4336	0.18404	281.82	815.13	79.83	894.96	1176.78
80	11,520	311.81	5.3697	0.18623	282.71	814.45	79.89	894.33	1177.04
81	11,664	312.67	5.3076	0.18841	283.59	813.76	79.95	893.71	1177.30
82	11,808	313.52	5.2470	0.19059	284.46	813.09	80.01	893.10	1177.56
83	11,952	314.36	5.1875	0.19277	285.33	812.42	80.07	892.49	1177.82
84	12,096	315.20	5.1295	0.19495	286.19	811.75	80.13	891.88	1178.07
85	12,240	316.02	5.0728	0.19713	287.04	811.10	80.19	891.29	1178.33
86	12,384	316.84	5.0173	0.19931	287.89	810.44	80.25	890.69	1178.58
87	12,528	317.65	4.9632	0.20148	288.71	809.81	80.30	890.11	1178.82
88	12,672	318.45	4.9104	0.20365	289.54	809.18	80.35	889.53	1179.07
89	12,816	319.25	4.8586	0.20582	290.36	808.55	80.40	888.95	1179.31
90	12,960	320.04	4.8079	0.20799	291.17	807.93	80.45	888.38	1179.55
91	13,104	320.82	4.7582	0.21016	291.98	807.31	80.50	887.81	1179.79
92	13,248	321.60	4.7099	0.21232	292 78	806.69	80.56	887.25	1180.03
93	13,392	322.37	4.6624	0.21448	293.57	806.08	80.61	886.69	1180.26
94	13,536	323.13	4.6160	0.21664	294.35	805.48	80.66	886.14	1180.49
95	13,680	323.88	4 5704	0.21880	295.13	804.88	80.71	885 59	1180.72
96	13,824	324.63	4.5259	0.22095	295.91	804.28	80.76	885.04	1180.95
97	13,968	325.38	4.4823	0.22310	296.67	803.70	80.81	884.51	1181.18
98	14,112	326.11	4.4395	0.22525	297.44	803.11	80.86	883.97	1181.41
99	14,256	326.85	4.3975	0.22740	298.19	802.53	80.91	883.44	1181.63
100	14,400	327.57	4.3563	0.22955	298.94	801.96	80.95	882.91	1181.85

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Pressure in lbs. per sq. in.	Pressure in lbs. per sq. ft.	Temperature in degrees F.	Volume of one lb. of pure saturated steam, in cubic feet.	Weight of one cubic foot of pure saturated steam, in lbs.	Heat of one lb. of saturated water, in thermal units.	Internal latent heat of one lb. of pure saturated steam, in thermal units.	External latent heat of one lb. of pure saturated steam, in thermal units.	Total latent heat of one lb. of pure saturated steam, in thermal units. $r = \rho + APu$.	Total heat of one lb. of saturated steam, in thermal units. $\lambda = q + r$. $\lambda = q + \rho + APu$.
p	P	t	v	δ	q	ρ	APu	r	λ
101	14,544	328.29	4.3159	0.23170	299.68	801.39	81.00	882.39	1182.07
102	14,688	329.01	4.2762	0.23385	300.42	800.82	81.05	881.87	1182.29
103	14,832	329.71	4.2373	0.23600	301.14	800.26	81.10	881.36	1182.50
104	14,976	330.42	4.1990	0.23815	301.87	799.71	81.14	880.85	1182.72
105	15,120	331.11	4.1615	0.24030	302.59	799.16	81.18	880.34	1182.93
106	15,264	331.81	4.1246	0.24245	303.30	798.61	81.23	879.84	1183.14
107	15,408	332.49	4.0883	0.24460	304.01	798.07	81.27	879.34	1183.35
108	15,552	333.17	4.0528	0.24674	304.72	797.53	81.31	878.84	1183.56
109	15,696	333.85	4.0183	0.24886	305.42	796.99	81.36	878.35	1183.77
110	15,840	334.52	3.9841	0.25100	306.10	796.46	81.41	877.87	1183.97
111	15,984	335.19	3.9504	0.25314	306.79	795.93	81.45	877.38	1184.17
112	16,128	335.85	3.9173	0.25528	307.48	795.40	81.50	876.90	1184.38
113	16,272	336.51	3.8847	0.25742	308.16	794.88	81.54	876.42	1184.58
114	16,416	337.17	3.8527	0.25956	308.84	794.36	81.58	875.94	1184.78
115	16,560	337.81	3.8212	0.26170	309.50	793.85	81.62	875.47	1184.97
116	16,704	338.46	3.7902	0.26384	310.17	793.34	81.66	875.00	1185.17
117	16,848	339.10	3.7597	0.26598	310.83	792.84	81.70	874.54	1185.37
118	16,992	339.74	3.7297	0.26812	311.49	792.33	81.74	874.07	1185.56
119	17,136	340.37	3.7003	0.27025	312.14	791.83	81.78	873.61	1185.75
120	17,280	341.00	3.6714	0.27238	312.78	791.34	81.82	873.16	1185.94
121	17,424	341.62	3.6429	0.27451	313.43	790.84	81.86	872.70	1186.13
122	17,568	342.24	3.6150	0.27663	314.07	790.35	81.90	872.25	1186.32
123	17,712	342.85	3.5875	0.27875	314.71	789.86	81.94	871.80	1186.51
124	17,856	343.47	3.5604	0.28087	315.34	789.38	81.98	871.36	1186.70
125	18,000	344.07	3.5337	0.28299	315.97	788.89	82.02	870.91	1186.88
126	18,144	344.68	3.5074	0.28511	316.60	788.41	82.06	870.47	1187.07
127	18,288	345.28	3.4815	0.28723	317.22	787.94	82.09	870.03	1187.25
128	18,432	345.88	3.4560	0.28935	317.83	787.47	82.13	869.60	1187.43
129	18,576	346.46	3.4309	0.29147	318.44	787.06	82.17	869.17	1187.61
130	18,720	347.06	3.4061	0.29359	319.05	786.53	82.21	868.74	1187.79
131	18,864	347.64	3.3817	0.29571	319.66	786.06	82.25	868.31	1187.97
132	19,008	348.23	3.3576	0.29783	320.27	785.60	82.28	867.88	1188.15
133	19,152	348.81	3.3339	0.29995	320.87	785.14	82.32	867.46	1188.33
134	19,296	349.38	3.3105	0.30207	321.46	784.69	82.35	867.04	1188.50
135	19,440	349.95	3.2874	0.30419	322.06	784.24	82.38	866.62	1188.68
136	19,584	350.52	3.2648	0.30630	322.64	783.79	82.42	866.21	1188.85
137	19,728	351.09	3.2424	0.30841	323.23	783.34	82.45	865.79	1189.02
138	19,872	351.65	3.2204	0.31052	323.82	782.89	82.49	865.38	1189.20
139	20,016	352.21	3.1987	0.31263	324.40	782.45	82.52	864.97	1189.37
140	20,160	352.77	3.1775	0.31474	324.97	782.01	82.56	864.57	1189.54
141	20,304	353.32	3.1561	0.31685	325.54	781.57	82.59	864.16	1189.70
142	20,448	353.87	3.1352	0.31896	326.11	781.13	82.63	863.76	1189.87
143	20,592	354.42	3.1146	0.32107	326.68	780.70	82.66	863.36	1190.04
144	20,736	354.96	3.0943	0.32318	327.24	780.27	82.69	862.96	1190.20
145	20,880	355.50	3.0742	0.32529	327.80	779.85	82.72	862.57	1190.37
146	21,024	356.04	3.0544	0.32739	328.36	779.42	82.75	862.17	1190.53
147	21,168	356.57	3.0350	0.32949	328.92	778.99	82.79	861.78	1190.70
148	21,312	357.11	3.0158	0.33159	329.47	778.57	82.82	861.39	1190.86
149	21,456	357.64	2.9968	0.33369	330.01	778.15	82.86	861.01	1191.02
150	21,600	358.16	2.9781	0.33579	330.56	777.73	82.89	860.62	1191.18
151	21,744	358.68	2.9600	0.33789	331.10	777.32	82.92	860.24	1191.34
152	21,888	359.20	2.9418	0.33998	331.64	776.91	82.95	859.86	1191.50
153	22,032	359.72	2.9234	0.34207	332.18	776.50	82.98	859.48	1191.66
154	22,176	360.24	2.9056	0.34416	332.71	776.09	83.01	859.10	1191.81
155	22,320	360.75	2.8882	0.34625	333.24	775.69	83.04	858.73	1191.97

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Pressure in lbs. per sq. in.	Pressure in lbs. per sq. ft.	Temperature in degrees F.	Volume of one lb. of pure saturated steam, in cubic feet.	Weight of one cubic foot of pure saturated steam, in lbs.	Heat of one lb. of saturated water, in thermal units.	Internal latent heat of one lb. of pure saturated steam, in thermal units.	External latent heat of one lb. of pure saturated steam, in thermal units.	Total latent heat of one lb. of pure saturated steam, in thermal units. $r = \rho + APu$.	Total heat of one lb. of saturated steam, in thermal units. $\lambda = q + r$, $\lambda = q + \rho + APu$.
p	P	t	v	δ	q	ρ	APu	r	λ
156	22,464	361.26	2.8708	0.34834	333.78	775.28	83.07	858.35	1192.13
157	22,608	361.77	2.8536	0.35043	334.30	774.88	83.10	857.98	1192.28
158	22,752	362.27	2.8367	0.35252	334.83	774.48	83.13	857.61	1192.44
159	22,896	362.78	2.8200	0.35461	335.35	774.08	83.16	857.24	1192.59
160	23,040	363.28	2.8034	0.35670	335.87	773.68	83.19	856.87	1192.74
161	23,184	363.77	2.7871	0.35879	336.38	773.29	83.22	856.51	1192.89
162	23,328	364.27	2.7710	0.36088	336.89	772.90	83.25	856.15	1193.04
163	23,472	364.76	2.7551	0.36296	337.41	772.50	83.28	855.78	1193.19
164	23,616	365.26	2.7394	0.36504	337.92	772.11	83.31	855.42	1193.34
165	23,760	365.74	2.7239	0.36712	338.42	771.73	83.34	855.07	1193.49
166	23,904	366.23	2.7085	0.36920	338.93	771.34	83.37	854.71	1193.64
167	24,048	366.72	2.6934	0.37128	339.44	770.96	83.39	854.35	1193.79
168	24,192	367.20	2.6784	0.37336	339.94	770.58	83.42	854.00	1193.94
169	24,336	367.68	2.6635	0.37544	340.43	770.20	83.45	853.65	1194.08
170	24,480	368.16	2.6489	0.37752	340.94	769.82	83.48	853.29	1194.23
171	24,624	368.63	2.6344	0.37960	341.42	769.44	83.51	852.95	1194.37
172	24,768	369.11	2.6200	0.38168	341.92	769.06	83.54	852.60	1194.52
173	24,912	369.58	2.6058	0.38376	342.41	768.69	83.56	852.25	1194.66
174	25,056	370.05	2.5917	0.38584	842.89	768.32	83.59	851.91	1194.80
175	25,200	370.51	2.5778	0.38792	343.38	767.95	83.62	851.57	1194.95
176	25,344	370.98	2.5641	0.39000	343.87	767.58	83.64	851.22	1195.09
177	25,488	371.44	2.5505	0.39208	344.35	767.21	83.67	850.88	1195.23
178	25,632	371.90	2.5370	0.39416	344.83	766.84	83.70	850.54	1195.37
179	25,776	372.36	2.5237	0.39624	345.30	766.48	83.73	850.21	1195.51
180	25,920	372.82	2.5105	0.39832	345.78	766.12	83.75	849.87	1195.65
181	26,064	373.28	2.4975	0.40040	346.26	765.76	83.77	849.53	1195.79
182	26,208	373.73	2.4846	0.40248	346.73	765.40	83.80	849.20	1195.93
183	26,352	374.18	2.4718	0.40456	347.20	765.04	83.83	848.87	1196.07
184	26,496	374.63	2.4592	0.40664	347.66	764.68	83.86	848.54	1196.20
185	26,640	375.08	2.4467	0.40872	348.13	764.33	83.88	848.21	1196.34
186	26,784	375.53	2.4343	0.41080	348.60	763.98	83.90	847.88	1196.48
187	26,928	375.97	2.4220	0.41288	349.06	763.63	83.92	847.55	1196.61
188	27,072	376.41	2.4099	0.41496	349.52	763.28	83.95	847.23	1196.75
189	27,216	376.85	2.3979	0.41704	349.97	762.94	83.97	846.91	1196.88
190	27,360	377.29	2.3860	0.41912	350.43	762.59	83.99	846.58	1197.01
191	27,504	377.73	2.3741	0.42122	350.88	762.25	84.02	846.27	1197.15
192	27,648	378.16	2.3623	0.42331	351.34	761.90	84.04	845.94	1197.28
193	27,792	378.59	2.3507	0.42540	351.79	761.56	84.06	845.62	1197.41
194	27,936	379.02	2.3392	0.42749	352.24	761.22	84.08	845.30	1197.54
195	28,080	379.45	2.3279	0.42957	352.69	760.88	84.10	844.98	1197.67
196	28,224	379.88	2.3167	0.43165	353.13	760.54	84.13	844.67	1197.80
197	28,368	380.31	2.3056	0.43373	353.57	760.20	84.16	844.36	1197.93
198	28,512	380.73	2.2946	0.43581	354.01	759.86	84.19	844.05	1198.06
199	28,656	381.15	2.2837	0.43788	354.45	759.53	84.21	843.74	1198.19
200	28,800	381.57	2.2730	0.43995	354.89	759.20	84.23	843.43	1198.32
201	28,944	381.99	2.2623	0.44202	355.33	758.86	84.26	843.12	1198.45
202	29,088	382.41	2.2518	0.44409	355.77	758.53	84.28	842.81	1198.58
203	29,232	382.83	2.2414	0.44615	356.20	758.20	84.30	842.50	1198.70
204	29,376	383.25	2.2311	0.44821	356.63	757.87	84.33	842.20	1198.83
205	29,520	383.66	2.2209	0.45027	357.06	757.55	84.35	841.90	1198.96
206	29,664	384.07	2.2108	0.45233	357.49	757.22	84.37	841.59	1199.08
207	29,808	384.48	2.2008	0.45438	357.92	756.89	84.40	841.29	1199.21
208	29,952	384.89	2.1909	0.45643	358.34	756.57	84.42	840.99	1199.33
209	30,096	385.29	2.1811	0.45848	358.76	756.25	84.44	840.69	1199.45

1	2	3	4	5	6	7	8	9	10
Pressure in lbs. per sq. in.	Pressure in lbs. per sq. ft.	Temperature in degrees F.	Volume of one lb. of pure saturated steam, in cubic feet.	Weight of one cubic foot of pure saturated steam, in lbs.	Heat of one lb. of saturated water, in thermal units.	Internal latent heat of one lb. of pure saturated steam, in thermal units.	External latent heat of one lb. of pure saturated steam, in thermal units.	Total latent heat of one lb. of pure saturated steam, in thermal units. $r = \rho + APu$.	Total heat of one lb. of saturated steam, in thermal units. $\lambda = q + r$, $\lambda = q + \rho + APu$.
p	P	t	v	δ	q	ρ	APu	r	λ
210	30.240	385.69	2.1714	0.46058	359.18	755.94	84.46	840.40	1199.58
211	30.384	386.09	2.1618	0.46257	359.60	755.62	84.48	840.10	1199.70
212	30.528	386.50	2.1523	0.46461	360.02	755.29	84.51	839.80	1199.82
213	30.672	386.90	2.1429	0.46665	360.44	754.97	84.53	839.50	1199.94
214	30.816	387.30	2.1336	0.46869	360.86	754.66	84.55	839.21	1200.07
215	30,960	387.69	2.1244	0.47072	361.27	754.35	84.57	838.92	1200.19
216	31.104	388.09	2.1153	0.47275	361.68	754.03	84.60	838.63	1200.31
217	31,248	388.48	2.1063	0.47477	362.09	753.72	84.62	838.34	1200.43
218	31.392	388.87	2.0974	0.47679	362.50	753.41	84.64	838.05	1200 55
219	31,536	389.26	2.0885	0.47881	362.91	753.09	84.66	837.75	1200.66

DISCUSSION.

Prof. J. E. Denton.—This paper sets forth the methods of studying the amounts of cylinder condensation and re-evaporation occurring at the several periods of the cycle of a steam engine, which have been made familiar to American students through the exhaustive digests of the investigations of Hirn, Hallauer, and others, contained in the excellent treatise on steam by the late Prof. Chas. A. Smith. Such studies are now quite general among American experimenters and teachers, but it is nevertheless very satisfactory to have the transactions of the Society honored by this contribution from so distinguished a worker as the writer of this paper. I believe that the graphical processes and the general character of the presentations of method are more complete than anything heretofore offered.

Prof. H. W. Spangler.—It is well known through the efforts of Prof. Thurston and other members of this society, that the condensation in the steam engine, which the writer's diagram purports to show, is caused primarily by the surface that is exposed to the exhaust and then to the entering steam, and in which the element of time comes in to a very great extent as well as the element of surface. In these diagrams, which are given, we have what seems to be a rectangular area showing the transmission

of heat during the time of the entering of steam, which would lead one to infer that the supposition of the author was that the transmission of heat was something which took place through a certain definite part of the stroke at a certain definite rate. Now the fact of the matter is, so far as we know, at the early part of the stroke, at the admission, the greater quantity of condensation must take place because the greater surface has been exposed for the greater length of time to the exhaust, and, it seems to me, as a graphical representation of what takes place in a steam cylinder, this falls considerably short of being correct. As to the advantage of these particular tables, those of our colleague, Mr. Peabody, seem to cover the case exactly for our use, and, I think, are more complete.

Prof. R. H. Thurston.—In regard to the tables, I understand that they are the tables which the author uses in making his computations. I do not understand him to say that they are better or worse than other and standard tables. He has reduced them to British units, from the metric system, for his own special purposes. I find in print an embarrassing number of steam tables, and would be much pleased if most of them were wiped out of existence and only those prepared by such experts retained.

In regard to the other questions, as to whether the paper shows the transfer and method of transfer, a glance at the paper will answer that question without any doubt. The paper apparently shows no attempt to exhibit the complete method of variation of this flow, but simply its total amounts, positive and negative. But M. Dwelshauvers has elsewhere attempted to trace that process, and to give quantitive results—in an approximate way perhaps, but for the first time, certainly in an encouraging manner—in the papers to which I have already referred. These he has lately published in the Transactions of the British Institution of Civil Engineers, and in the bulletin of the Society of Mulhouse. He has made a first attempt to trace this transfer of heat between the steam and the steam-engine cylinder without as much success as he himself would have desired, or as much success as any of us would be glad to attain; but he has shown that it is perfectly practicable to adopt a certain method as a beginning, which method is likely to prove to be valuable. We have to thank him, not for having done the work completely, but for having shown how to go about the beginning of it. I do not think we have, in any one direction, as yet reached the end. We

are making attempts, some more, some less successful, to secure a knowledge of the method of loss of heat and of steam in the engine cylinder, as ordinarily constructed, by this kind of transfer. I think that we are under great obligations to M. Dwelshauvers for going even as far as he has gone. He states that he has not been able to secure an accurate measure of the quantity and rate of such flow at all points in the engine cycle. He hopes that, in time, a way may be found; but he as yet has not been able to obtain results which are entirely satisfactory to himself. He has taken most carefully made indicator cards and has endeavored to work from them, and with some success; but it is a success at which we are all pleased as the beginning, but not the success at which we would be still better pleased were it the end.

CCCLX.

COST OF STEAM AND WATER POWER.

BY CHAS. T. MAIN, LAWRENCE, MASS.

(Member of the Society.)

During the past two or three years the interest taken in the relative cost of steam and water power has shown itself in frequent discussions and by a few papers on the subject. These papers are "The Statistics of the Water Power Employed in Manufacturing in the United States," by Professor Swain, presented to the American Statistical Association and printed in their publication of March, 1888; a paper by Mr. Chas. H. Manning, "Comparative Cost of Steam and Water Power," read at the last meeting of this Society; * and a paper by the writer presented in 1886 to the students at the Massachusetts Institute of Technology. Since 1886 I have been able to collect a very much larger amount of data bearing upon this subject, and, without at all criticising the other papers published, will now put a part of such data into definite form for further information on this subject.

The costs of power which follow have been worked out per indicated horse power on engine, and horse power on wheel shaft. These two are comparable, for the friction of engine will be about equal to the friction of gears and shafting on water wheels.

I. Let us first consider the cost of steam power, and we shall find that its cost wiil vary through a number of causes, which are principally the cost of fuel in different localities; the amount of exhaust steam used for various purposes after passing it through the engine and there producing power; the amount of power required and the type of engine used.

In order to cover all the variable quantities in our problem, Table I. has been prepared, which shows the approximate cost of steam power in amounts varying from 500 to 2,000 H. P. with the high pressure, condensing, and compound engine, using different per cents. of exhaust steam for heating purposes, and various prices of coal.

* Trans. A. S. M. E.: Vol. X., p. 499, No. CCCXXXIX.

The methods used in working out this table are fully explained by working through for a 1,000 H.P. plant, using an average of 25% of exhaust steam for heating purposes, with coal at $5.00 per long ton delivered in pocket.

Let us assume, in the plant of 1,000 I. H. P., that the engines for each case are a pair of tandem compounds, a pair of single cylinder condensing, and a pair of non-condensing engines.

The items of cost for running such plants are fuel, attendance of engines and boilers, oil, waste, and supplies, depreciation, repairs, interest, taxation, and insurance.

The fuel consumption per indicated horse power per hour is shown in the following table. The total consumption per I. H. P. per hour is the total amount burned, and is the amount to be charged to power if no exhaust steam is used for heating purposes. The net consumption per I. H. P. per hour is the amount to be charged to power after deducting a weight equivalent to the amount of exhaust steam used for heating purposes.

The conditions for running are assumed as follows: The compound engine is to run with 100 lbs. initial pressure above the atmosphere; the receiver pressure to be 5 lbs. The condensing engine to have an initial pressure of 80 lbs.; and if a portion is run high-pressure, that portion is to exhaust against 5 lbs. back pressure. The high-pressure engine to run with an initial pressure of 100 lbs., and to exhaust against a back pressure of 5 lbs. The temperature of feed water is taken at 100° Fahr.

The coal consumption per I. H. P. per hour, when the engines are running with steam used for power only, are taken as 1¾ lbs. for the compound, 2½ lbs. for the condensing, and three lbs. for the high-pressure engines, and these figures will be conceded by engineers in general to be fair values to work with for the best constructed engines and boilers for each type.

TABLE I.—SHOWING APPROXIMATE TOTAL YEARLY EXPENSE OF STEAM POWER PER I. H. P.

Engine.	I. H. P. of Plant.	STEAM USED FOR POWER ONLY				25% OF EXHAUST STEAM USED.				50% OF EXHAUST STEAM USED.				75% OF EXHAUST STEAM USED.				100% OF EXHAUST STEAM USED			
		Cost of Coal per Ton.				Cost of Coal per Ton.				Cost of Coal per Ton.				Cost of Coal per Ton.				Cost of Coal per Ton.			
		$ 3.00	$ 4.00	$ 5.00	$ 6.00	$ 3.00	$ 4.00	$ 5.00	$ 6.00	$ 3.00	$ 4.00	$ 5.00	$ 6.00	$ 3.00	$ 4.00	$ 5.00	$ 6.00	$ 3.00	$ 4.00	$ 5.00	$ 6.00
Compound.	500	22.35	24.82	27.28	29.74	20.73	22.84	24.95	27.06	19.10	20.86	22.62	24.38	17.42	18.83	20.24	21.65				
	1000	19.16	21.63	24.09	26.55	17.57	19.68	21.79	23.90	15.99	17.75	19.51	21.27	14.34	15.75	17.16	18.57				
	1500	17.38	19.84	22.30	24.76	15.81	17.92	20.03	22.14	14.23	15.99	17.75	19.51	12.62	14.03	15.44	16.85				
	2000	16.23	18.69	21.15	23.61	14.73	16.84	18.95	21.06	13.18	14.94	16.70	18.46	11.66	13.07	14.48	15.89				
Condensing.	500	24.83	28.35	31.87	35.39	22.03	24.93	27.83	30.73	19.06	21.86	23.66	25.96	16.28	17.96	19.64	21.32				
	1000	22.34	25.84	29.36	32.88	19.54	22.44	25.34	28.24	16.62	18.92	21.22	23.52	13.87	15.54	17.22	18.90				
	1500	21.01	24.53	28.05	31.57	18.81	21.71	24.61	27.51	15.28	17.58	19.88	22.18	12.48	14.16	15.84	17.52				
	2000	20.13	23.65	27.17	30.69	17.93	20.83	23.73	26.63	14.40	16.70	19.00	21.30	11.60	13.28	14.96	16.64				
High-Pressure.	500	27.16	31.39	35.62	39.85	23.64	27.06	30.50	33.94	20.07	22.72	25.37	28.02	16.47	18.32	20.17	23.02	12.89	13.95	15.01	16.07
	1000	24.82	29.03	33.25	37.48	21.26	24.70	28.14	31.58	17.72	20.37	23.02	25.67	4.16	16.01	17.86	19.71	10.54	11.60	12.66	13.72
	1500	23.69	27.92	32.15	36.38	20.16	23.60	27.04	30.48	16.59	19.24	21.89	24.54	3.06	14.91	16.76	18.61	9.38	10.44	11.50	12.56
	2000	22.79	27.02	31.25	35.48	19.26	22.70	26.14	29.58	15.69	18.34	20.99	23.64	2.16	14.01	15.86	17.71	8.48	9.54	10.60	11.66

TABLE II.

SHOWING GROSS AND NET COAL CONSUMPTION IN LBS. PER INDICATED HORSE POWER PER HOUR.

Column 1.	2	3	4	5	6	7
ENGINE.	COMPOUND.		CONDENSING.		HIGH-PRESSURE.	
Per cent. of exhaust steam used for heating purposes.	Lbs. of coal per I. H. P. per hour, all coal charged to power.	Lbs. of coal per I. H. P. per hour, deducting amount equivalent to exhaust steam used.	Lbs. of coal per I. H. P. per hour, all coal charged to power.	Lbs. of coal per I. H. P. per hour, deducting amount equivalent to exhaust steam used.	Lbs. of coal per I. H. P. per hour, all coal charged to power.	Lbs. of coal per I. H. P. per hour, deducting amount equivalent to exhaust steam used.
0	1.75	1.75	2.50	2.50	3.00	3.00
25	2.06	1.50	2.63	2.06		2.44
50	2.38	1.25	2.75	1.63		1.88
75	2.69	1.00	2.88	1.19		1.31
100						0.75

Explanation of Table II.

When steam is used for power only, the coal consumption is shown at the head of each column.

In col. 2 is given the gross consumption per I. H. P. for the compound engine, which is found thus: Supposing 25% of the steam exhausted from the high-pressure cylinder is taken from the receiver, then 75% of the whole steam admitted to high-pressure cylinder will be used as in a regular compound, and 25% of the steam admitted will be used as in a high-pressure engine, and the total consumption will be

$$\begin{aligned} 1.75 \times 75 &= 1.31 \\ 3.00 \times .25 &= \ \ .75 \\ \hline &\ 2.06 \end{aligned}$$

and so on for each per cent. taken from receiver.

Col. 4 is obtained in a similar way to col. 2.

In col. 3 is given the net weight to charge to power. For the compound engine it is obtained thus: Starting at 100° temperature of feed, the amount of heat necessary to make one pound of steam at 100 lbs. will be 1,117 thermal units.

The amount of heat necessary to produce one pound of steam at 5 lbs. pressure from 100° Fahr. is 1,083 thermal units.

The amount of heat to charge to power if the steam is admitted

at 100 lbs. pressure, and exhausted and used at 5 lbs. pressure, will be $1{,}117 - 1{,}083 = 34$ T. U. per lb. of steam, or $34 \div 1{,}117 = .0307$ of the total amount admitted.

Besides the change of about 3%, due to the difference in pressure at the beginning and end of stroke, we must consider the effect of cylinder condensation, and condensation in the jackets. These are rather indefinite quantities, and will seriously affect the amount of coal to be charged to power. The difference in the amount of condensation by passing the steam through an engine, or passing it through pressure regulators and pipes, should be charged to the power. Let us assume that 20% of the steam apparently evaporated passes from each cylinder in the form of water, and that 5% of the total weight of steam used is condensed in the jackets, making a total loss by condensation of 25%. This may be rather high, but I have placed it high enough to cover possible losses by condensation. This added to 3%, the amount of heat due to the difference in pressures, makes a total loss of 28%. If we call the loss by condensation in pipes and passing by through regulators 3%, we shall have a difference or net charge of 25% to make to the engine or to power.

We get, then, the amount of steam to charge to power for the compound engine, 25% being taken from the receiver; thus

$$\begin{aligned} 1.75 \times .75 &= 1.3125 \\ 3.00 \times .25 \times .25 &= \underline{\ .1875} \\ &\ 1.50 \end{aligned}$$

and so on for each per cent. taken from the receiver.

Cols. 3, 5, 6 and 8 are obtained in a similar way.

If it is necessary to make steam for other purposes than power, and if this steam can be passed through the engine before being used for other purposes, then all the expense that should be charged to power is the weights of coal which we have in columns 3, 5, 6 and 8, Table I., a portion of the attendance on boilers, and a portion of the cost of boiler plant for depreciation, repairs, interest, taxation and insurance, this portion being in the same ratio to the full cost of attendance and depreciation, repairs, etc., of the boiler plant as the weight of coal charged to the engine is to the gross weight consumed. To this add the full cost of attendance of engine, full cost of oil, waste, and supplies for engine, and full cost of engine plant for depreciation, repairs, etc.

This is shown in Tables III. and IV.

TABLE III.—SHOWING ORDINARY RUNNING DAILY AND YEARLY EXPENSES. 1,000 H. P. PLANT.

Col. 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
Per cent. of exhaust steam used.	Lbs. coal per I.H.P. per hour.			Cost of coal per I.H.P. per day of 10¼ hours @ $5.00 per long ton—2,240 lbs.			Atten'ance of boilers per I.H.P. per day.			Attendance of engine per I.H.P. per day.			Oil, waste & supplies per I.H.P. per day.			Total daily expense.			Total yearly expense —308 days.		
	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g.	H. P.	Compound.	Condens'g	H. P.	Compound.	Condens'g.	H. P.
				c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	c.	$	$	$
0	1.75	2.50	3.00	4.00	5.72	6.86	0.53	0.75	0.90	0.60	0.40	0.35	0.25	0.22	0.20	5.38	7.09	8.31	16.570	21.837	25.595
25	1.50	2.06	2.44	3.43	4.71	5.58	.45	.62	.73	.60	.40	.35	.25	.22	.20	4.73	5.95	6.86	14.568	18.326	21.129
50	1.25	1.63	1.88	2.86	3.73	4.30	.38	.49	.56	.60	.40	.35	.25	.22	.20	4.09	4.84	5.41	12.597	14.907	16.663
75	1.00	1.19	1.31	2.29	2.72	3.00	.30	.36	.39	.60	.40	.35	.25	.22	.20	3.43	3.72	3.94	10.564	11.458	12.135
100	......	......	0.75	......	......	1.72	......	......	.23	......	......	.35	......	......	.20	......	......	2.50	......	......	7.700

TABLE IV.—SHOWING COST OF PLANT PER I. H. P. TO CHARGE TO POWER. ALSO DEPRECIATION, REPAIRS, INTEREST, TAXATION AND INSURANCE PER I. H. P. PER YEAR. ALSO TOTAL YEARLY EXPENSE PER I. H. P.

Col. 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Engine.	Per cent. of exhaust steam used.	Engine and piping complete.	Engine house.	Engine foundations.	Total cost of engine plant.	Depreciation at 4 % on total cost.	Repairs @ 2 % on total cost.	Interest @ 5 % on total cost.	Taxation @ 1.5 % on ⅗ cost.	Insurance @ 0.5 % on engine and engine house.	Totals of Cols. 7, 8, 9, 10, and 11.	Boilers complete, including feed pumps, etc.	Boiler house.	Chimney and flues.	Total cost of boiler plant.	Depreciation @ 5 % on total cost.	Repairs @ 2 % on total cost.	Interest @ 5 % on total cost.	Taxation @ 1½ % on ⅗ cost.	Insurance at 0.5 % on total cost.	Totals of Cols. 17, 18, 19, 20, and 21.	Total ordinary running expense Cols. 20, 21, and 22, Table II.	Total yearly expense per I. H. P. Cols. 12 + 22 + 23.
		$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$
Compound.	0	25.00	8.00	7.00	40.00	1.60	0.80	2.00	0.45	0.165	5.015	9.33	2.92	6.11	18.36	0.918	0.367	0.918	0.207	0.092	2 502	16.570	24.087
	25	Constant.	"	"	"	"	"	"	"	"	"	8.00	2.50	5.66	16.16	.808	.323	.808	.182	.081	2.202	14.568	21 785
	50											6 67	2.08	5.15	13.90	.695	.278	.695	.156	.070	1.894	12.597	19.506
	75											5.33	1.67	4.60	11.60	.580	.232	.580	.131	.058	1.581	10.564	17.160
Condensing.	0	20.00	7.50	5.50	33.00	1.32	0.66	1.65	0.371	0.188	4.139	13.33	4.17	7.30	24.80	1.240	.496	1.240	.279	.124	3.379	21.837	29.355
	50	20.00	7.50	5.50	33.00	1.32	0.66	1.65	.371	.133	4.139	10.99	3.43	6.70	21.12	1.056	.422	1.056	.238	.106	2.878	18.326	25.343
	25	19.00	7.50	5.00	31.00	1.26	0.63	1.575	.354	133	3.952	8.69	2.72	5.92	17.33	.867	.347	.867	.195	.086	2.3 2	14.907	21.221
	75	19.00	7.50	5.00	31.00	1.26	0.63	1.575	.354	.133	3.952	6.35	1.98	4.92	13.25	.663	.265	.663	.149	.066	1.806	11.458	17.216
Non-Condening.	0	17.50	7.50	4.50	29.50	1.18	0.59	1.475	0.332	0.125	3.702	16.00	5.00	8.00	29.00	1.450	.580	1.450	.326	.145	3.951	25.595	33.248
	50	Constant.	"	"	"	"	"	"	"	"	"	13.01	4.07	7.20	24.28	1.214	.486	1.214	.273	.121	3.308	21.129	28.139
	25											10.03	3.13	6.30	19 46	.973	.392	.973	.219	.097	2.654	16.663	23.019
	75											6.99	2.18	5 30	14.47	.724	.289	.724	.163	.072	1.972	12.185	17.859
	100											4.00	1.25	4.00	9.25	.463	.185	.463	.104	.046	1.261	7.700	12.663

II. The variation in the cost of water power plant per horse power is very great. This variation is caused principally by the following reasons: First, the variation in head in different localities; and unless the fall is between two canals, the head is variable for a specified place, and therefore influences the size of plant necessary. Second, the distance from source of supply to point of discharge is an element which very materially affects the cost of plant.

The variation in total cost of water power will depend then principally upon the charge made for water and the cost of plant per unit of power.

The average estimated cost per H. P. on wheel shaft for three separate plants in Manchester, N. H., of 890 H. P. each, and very nearly identical in construction, where the average distance from canal to river is about 330 feet, and the average head 30 feet, is $44. As these all discharge into the river, they must either be about one and a half times the capacity required at average head, or a portion of the mill will be stopped many days during the year, in times of freshets when the head is reduced, and some days even if the capacity is at average head one and one-half times the average power required. The cost then of these plants will be $\$44 \times 1\frac{1}{2} = \$66$ per H. P. required.

In Lowell, Mass., in one mill, there are two plants where the average distance from supply canal to discharge canal is about 575 feet, and the fall 13 feet. The estimated cost per H. P. for a modern plant is about $110. The power of each plant would be about 1,000 H. P. As the discharge is into a canal, the head is nearly uniform throughout the year.

On another corporation in Lowell there are four distinct plants, all delivering from canal to river under an average head of 18 feet. The average distance from canal to river is about 290 feet. The estimated costs of four modern plants under the same conditions average about $57 per H. P. All of these are subject to back water, and the plants must be of one and a half times the power required, and, in fact, the plant at this corporation is two-thirds greater capacity at average head than the average power used.

The estimated cost of one plant in Lawrence, Mass., where the average head is 28 feet and the distance from canal to river 400 feet, is $42 per H. P. of plant, or $\$42 \times 1\frac{1}{2} = \$63$ per H. P. used.

In 1887 there were twenty-five days when the wheels would develop only two-thirds or under of the power at 28 feet head in

Lawrence, and for year ending May 1, 1888, there were thirty-eight days at Lowell when some of the wheels were running under similar conditions. During the year 1888 the number of days when the power was greatly reduced was even more than in 1887.

From the above it will be seen that no established cost can be made for a water power plant, even in the same place when the conditions differ, and that the cost per H. P. on wheel shaft may easily vary from $50 to $100, and therefore that each case must be treated by itself.

The items which go to make up the total cost of a plant in Lawrence, North Canal, for a 1,000 H. P. plant, are about as follows:—

Ice guards, racks, and head gates	$ 3.50
Iron penstocks	8.50
Wheels (flume pattern), including gears and shafting	10.00
Wheel pits	8.00
Raceways	15.00
	$45.00

$\$45 \times 1\frac{1}{2} = \$67.50$ per H. P. used.

Table V. shows the approximate cost per H. P. on wheel shaft for a 1,000 H. P. plant under different heads, and different distances from supply to discharge, where no great obstacles to construction exist.

TABLE V.

Head in Feet.	Distance from Supply to Discharge in Feet.					
	100	200	300	400	500	600
10	$95	$110	$125	$140	$155	$170
20	38	46	54	62	70	77
30	26	32	39	45	51	58
40	20	25	31	37	42	48
50	19	23	28	33	37	41

The above table and other estimated costs given do not include the value of building or portion of building used for wheel-room, which would be about $2 per H. P. for head of 30 feet.

To get the cost of a plant of any other size than 1,000 H. P. add one per cent. to cost in table for each 100 H. P. decrease, and subtract one per cent. for each 100 H. P. increase. This holds good between the limits of 500 and 2,000 H. P.

If the discharge is into river, the plant will necessarily be greater than that required at average head, and the figures in the table must be increased by the per cent. of reserve power necessary.

The various charges for water in Manchester, Lowell, and Lawrence are as follows:

Manchester.

About \$300 per year per mill power for original purchases.
\$2 per day per mill power for surplus.

Lowell.

About \$300 per year per mill power for original purchases.
\$2 per day per mill power during "back-water."
\$4 per day per mill power for surplus under 40 per cent.
\$10 per day per mill power for surplus over 40 per cent. and under 50 per cent.
\$20 per day per mill power for surplus over 50 per cent.
\$75 per day per mill power for any excess over limitation.

Lawrence.

About \$300 per year per mill power for original purchases.
About \$1,200 per year per mill power for new leases at present.
\$4 per day per mill power for surplus up to 20 per cent.
\$8 per day per mill power for surplus over 20 and under 50 per cent.
\$4 per day per mill power for surplus over 50 per cent.

One mill power in Lawrence = 30 cubic feet of water falling through 25 feet head per second. The horse power of the water is due to a head of about one foot less than 25, to allow for the loss in getting the water on and off the wheel, so that we have

$$1\ M.\ P. = \frac{30 \times 24 \times 62.3}{550} = 81.56\ H.\ P.$$

With wheels of 80 per cent. efficiency the power on wheel shaft equals 65.25 H. P.

A mill power in Lowell is equivalent to that in Lawrence.

In Manchester a mill power = 38 cubic feet per second on a fall of 20 feet. Allowing one foot loss we have

$$\frac{38 \times 19 \times 62.3}{550} = 81.78\ H.\ P.$$

With 80 per cent. efficiency on wheels, this equals 65.42 H. P. on wheel shaft.

The H. P. per M. P. is near enough to call it the same for these three places, 65 H. P. The net effective horse power would be about 60.

Attendance, one man at \$2.00, for 308 days	\$616
Oil, waste, and supplies	100
	\$716

or about \$0.72 per H. P. per year for a 1,000 H. P. plant.

TABLE VI.—Showing Yearly Expense of Water Power per H. P. on Wheel Shaft.

Charges for Water.		Attendance, oil, supplies, etc.	Fixed Charges on Cost of Plant. Cost of plant.						Total Yearly Expense per H. P. For plants costing					
Per mill power.	Per H.P. per year.		$50	$60	$70	$80	$90	$100	$50	$60	$70	$80	$90	$100
$300 per year ... a	$4.62	$0.72	$5.08	$6.10	$7.11	$8.12	$9.13	$10.15	$10.42	$11.44	$12.45	$13.46	$14.47	$15.49
$300 per year ... b	12.31								18.11	19.13	20.14	21.15	22.16	23.18
2 per day.......	9.48								15.28	16.30	17.31	18.32	19.33	20.35
4 " "	18.96								24.76	25.78	26.79	27.80	28.81	29.83
8 " "	37.92								43.72	44.74	45.75	46.76	47.77	48.79
10 " "	47.40								53.20	54.22	55.23	56.24	57.25	58.27
20 " "	94.80								100.60	101.62	102.63	103.64	104.65	105.67

Explanation of Table.

a, in column "Per H. P. per year," is the cost, not including interest, on the original purchase.

b, in column "Per H. P. per year," is the cost, including interest, on original purchase, which amounts to about $7.49 per H. P., or 5 per cent. on $10,000 per M. P. per year.

The "Fixed Charges on Cost of Plant" are depreciation at 2.5 per cent. average, repairs at 1½ per cent., interest at 5 per cent., taxes at 1½ per cent. in ¾ cost, and insurance at .05 per cent. on exposed portion.

III. From a comparison of Tables I. and VI., selecting the conditions from each which suit the case under consideration, can be obtained the relative cost, pure and simple, of steam and water power.

There is no doubt but what the great water powers of New England have caused the large manufacturing cities to be established and developed where they have been. At the time of their starting and development the steam engine had not reached such high efficiency as is now obtained. Immense sums of money were expended in some places for the development of the water power and the purchase of rights to use this power. The question is frequently asked, If steam power is cheaper than water power in many cases, why do you continue to use water power? The answer to this question is: Having spent this money, and now having the plant and privilege paid for, it is usually cheaper to use the water power than to put in steam power.

This question should be modified to read: Could you afford, at the present day, to pay such sums as would be required in the purchase of rights to use, and construction of plant for, water power? This question can be answered only by careful estimates of cost which have been approximately shown in the tables.

Thus far we have considered simply the *cost* of power. There may be advantages or necessities attending the use of either steam or water which might offset any difference in cost of power simply.

Some of the advantages which are favorable to steam power are a steady power throughout the year, independent of the rise and fall of a river; more uniform speed than can be obtained from water wheels, which are less sensitive in their regulation than an engine properly regulated; choice in selection of the site with reference to the markets, low freights, cheap fuel, favorable conditions for building, and procuring and keeping operatives.

Some of the advantages of water power over steam power are smaller amount of space occupied, which might be made available for other purposes, and greater cleanliness. In some cases large amounts of water are required for other purposes than power, and in such cases a location on a stream is desirable.

That the balance of advantages and cost combined is in favor of steam power for textile manufactures is proven at the present time by the erection of steam mills almost entirely, while there are still undeveloped water powers which are available.

In the "Analysis" of Vol. II., Census of Massachusetts, 1885, page cxix., is the following: "It is undoubtedly true that steam is taking the place of water as a motive power, especially in new enterprises. Steam power is also put in for use in case of the failure of the water supply."

DISCUSSION.

Mr. Samuel Webber.—I desire to offer as a part of the discussion on Mr. Mains' paper, the following notes which were prepared as a short independent paper, to correct some misapprehensions of Mr. Manning's as to the *cost* of water power. As far as my experience goes, I can agree with him very fully in regard to the cost of steam power, taking his case No. 2 as an example, but not in regard to the cost of water, in which he assumes the Essex Company's charge for *extra* water as a basis.

If such *charges* are taken as a basis for the cost of water power, they should be compared with the *charges* made for steam power by lessors in Boston or Worcester, which range all the way from $50 to $100 per H. P.

I have given in the sequel my views as to the *cost* of water power, quoting Mr. Nathan Appleton for the original rates fixed at Lowell, which were closely followed by all the other large water power companies in New England, but I omitted one very important statement made by Mr. Appleton before he mentions these rates, *viz.*, that "the first expenditure at Lowell for canals, etc., was $120,000, which was estimated to produce fifty mill powers of sixty net H. P. each."

This would be a cost of $2,400 per mill power, so that the rental fixed on this of $300 per year per M. P., would be 12½ per cent. on the cost, which shows that this was the total charge for the *water*, and that the payment for the land, as I say hereafter, should not be included. With the old "breast wheels" giving about 70 per cent. this was called 60 H. P., and would be $5 per annum per H. P. With the modern turbine, giving 80 per cent., it would yield 68 H. P. at $4.40 per H. P., which I assume as the total charge for water.

Mr. Denton asked, in the discussion on Mr. Manning's paper, if any allowance was made for repair of dams; and I can say that the dams at Lowell and Manchester have been rebuilt by the Water Power Companies in dressed stone and cement, like the one at Lawrence, and are likely to last for centuries.

Also, that the Holyoke Water Power Company made no reduction in charges for water due to any change in ownership. The original Lowell basis has always been the standard for all, until the over-use of water necessitated the *penal* charges spoken of.

I offer my views on the cost of water power gained by many years' experience, and commencing during the period of a residence in Lowell from 1841 to 1847, before any measurements of the water actually consumed by the mills had been made, and having witnessed the first of such experiments.

I cannot agree with Mr. Manning * that there should be charged to the cost of water power the sums paid by the different manufacturing companies for their mill-sites and land, including store-house and office sites, room for repair shops, coal sheds, etc., etc.

At the foundation of Lowell the price for these was fixed as follows, quoting from Mr. Nathan Appleton, one of the founders, in his pamphlet on "The Origin of the Power Loom," and of Lowell.

Mr. Appleton says: "The second mill built at Waltham contained 3,584 spindles, with all the apparatus necessary to spin No. 14 yarn and convert it into cloth, which was taken as a standard, and the necessary water power was estimated and established as the right to draw twenty-five cubic feet per second on a fall of thirty feet, or a gross H. P. of 85.05 supposed to net about 60 H. P. The price for this was fixed at Lowell at $4 per spindle, or $14,336 for a mill power and the necessary land, of which $5,000 was to remain unpaid, subject to an annual rent of $300 or $5 per H. P."

Now I have always understood, for nearly fifty years, that the sum paid down was for the necessary land, amounting to a number of acres more or less, and that the sum reserved, paying annual interest, was the cost of the water power; and I think I am corroborated in this opinion by the fact that the $300 per annum was the interest at 6 per cent. of $83.33 per H. P., which was more than the first cost of development, although that has been increased of later years by new dams and larger canals, which, however, furnish much more water for a larger portion of the year than the 10,000 H. P. originally estimated.

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* Transactions, Vol. X., p. 499, No. CCCXXXII.

The total capital of the Locks and Canals Company was only $600,000, and this covered all their purchases of land, besides grading streets and building a large machine shop; and this makes the total cost to them only $60 per H. P., so that the $5 rental may safely be considered as having covered the cost to the mills of their water.

The Essex Company's dam at Lawrence cost $250,000 for 10,000 H. P., or $25 per H. P., and though I do not *know*, I feel quite sure that the canal did not double this sum.

The new dam at Manchester, N. H., built in 1873 of dressed stone and cement, in the most thorough manner, cost $60,000, or $6 per H. P. for 10,000 H. P.

It is a rather singular coincidence that the available power on these three falls on the Merrimac river should be nearly identical—the fall at Manchester being 49 feet, with 2,590 miles drainage area, at 55 c. ft. per min. per mile, that at Lowell being 33 ft., with 4,000 miles drainage area, and at Lawrence 28 ft., with 4,600 miles area.

From the basis fixed for Lowell those of Manchester and Lawrence were established, so as practically to agree, and the odd foot of allowance at Manchester, which Mr. Manning speaks of, is a relic of the old days of breast wheels, which are now obsolete.

Having gone down the river from Lowell to Lawrence, to help in the early engineering of that city, and afterward lived a dozen years in Manchester, I feel pretty well conversant with these matters. No charge for extra water was made at Manchester until after 1871, when, the power being apparently exhausted, I was employed by the late Governor Straw, then agent of the Amoskeag Company, to make dynamometer measurements of the power consumed by the different mills, so as to restrict them to the amount they actually paid for, and use the surplus to build new mills on unsold and unoccupied land.

In this way the charges at Lawrence were fixed by the Essex Company so as to *penalize*, if possible, the use of extra water, and enable the company to utilize unsold land on the other side of the river.

Neither these extra charges nor the first cost of land belong to the real cost of water power, which was amply covered by the original charge of $300 per year per mill power, and on this I make up the following estimate :

Cost of water by turbine wheels, giving 80 per cent	$4 40
Sinking fund, etc., Manning	7 90
Attendance and oil	72
Total	$13 02

And I call this in excess, because I do not think it is necessary to add 33 per cent. to cost of plant.

The cost in many of the smaller powers in New England has not exceeded $100 per H. P. for dams, canals, wheels, and pits, and the total cost chargeable in such cases would not exceed $10 per H. P., while steam would be far higher.

With regard to the cost of steam, I see no reason to doubt the correctness of Mr. Manning's estimate of $14.58 per H. P. where all the exhaust is used for heating water for dyeing and bleaching, but I am skeptical as to the allowance of only $1\frac{3}{4}$ lbs. of coal per H. P. per hour, if any of the steam is taken from the engine for those purposes. I should prefer to estimate as follows:

Coal	$12 25
Attendance	2 44
Supplies	23
Sinking fund, etc	7 08
	$22 00
Plus 10 % for *net* power, as with water	2 20
	$24 20

for the total, and this is not quite half of what I have found the actual annual expenses per H. P. in some large cotton mills in New England, within 10 or 15 years, where the results were taken from the books, and not from 10-hour trials with selected fuel and firemen. I think Mr. Manning's estimates of cost of plant, for both water and steam, as close as possible.

I would simply quote the fact, to close, that some of the mills at Lowell prefer to pay $20 per annum for extra water, rather than resort to steam with their present plants, and that these payments for extra water, and other rentals, enable the Canal Company to supply the owners of original rights with water free of any charges whatever.

*Mr. Chas. T. Main.**—In answer to the contributed remarks, which were made really on Mr. Manning's paper and only in anticipation of my own, I will take up each point to which ex-

* Author's Closure, under the Rules.

ception is taken in the same order as presented, as nearly as possible.

I fail to see why the charges for extra water, above what is owned, should not be taken as a basis for the cost of power for that surplus water, for the users are obliged to pay in hard cash for every mill power or fraction thereof according to the rates stated in the paper, and if this surplus water can be offset by steam power, the cost of that steam power to set against it is the cost at that very place and nowhere else. That the charge for extra water was not a penal one in the case of the mill with which I have been connected for nine years, I am sure, for the Essex Company was very glad to let us use this water at the rates charged, and very sorry to have us dispense with the surplus in favor of steam power when we had arranged our plant to do so. Furthermore, in Lawrence, on the South Canal, *regular* power is now leased for $4 per mill power per day, or $1,200 per year, and is therefore a basis for the cost of water power in South Lawrence, which can be compared with the cost of steam power in South Lawrence, and *not* what steam power can be leased for in Boston or Worcester.

The original purchases of water power in Lawrence were made at $15,000 per mill power. $10,000 of this was paid down and $5,000 remained as perpetual interest at 6 per cent., and it is true that the land was practically given in with the purchase of the mill powers. If it were as Mr. Webber says, that the first payment, which in Lawrence amounted to $10,000 per mill power, was paid for land only, and that the $5,000 per mill power which remained as interest was the only payment for the water power, then the extravagant price of $250,000 was paid by the Pacific Mills for its upper site for land only, and $125,000 remained on interest at six per cent. in payment for the water power. As the mill site is subject to this perpetual rental for water power, whether the water is used or not, it seems as though the two were inseparable, and so they are usually considered, and the land would be of no value with the restrictions which are placed upon it, except for manufacturing purposes.

If the allowance of one foot from the gross head, which is made, is a relic of the old days of breast wheels, it is still found necessary to make it in Lawrence as well as Manchester, and I have found, by observing the heights of water at the canal and river, also at the wheel in the feeder and race-way, that the

head acting upon the wheel is just about one foot less than the difference between the canal and river, and this with fair-sized feeders and race-ways.

The necessity of adding 33%, as Mr. Manning says, or 50% as I claim, to the power of the plant above the average power required, it seems to me is clearly shown in the paper. Would it be better to shut down a portion of the work during the high water period than to put 50% extra cost into the water power plant?

I do not think that anybody will claim that if a compound engine can be run on 1¾ pounds of coal per H. P. per hour when no steam is taken from the receiver for heating purposes, that the consumption will be no more when steam is taken for such purposes. The consumption will increase from the minimum when no steam is taken out to the maximum of a high-pressure engine when all the exhaust steam is used.

In closing, I would say that I have—and I think Mr. Manning has also—considered this question according to the conditions which exist at the present time, without reference to what they may have been in the past. If we go back many years we shall certainly come to the time when there was no question of steam power being a competitor of water power under ordinary conditions, but now a very different set of conditions enters into the problem, and it is because of this change which has come about that we are able to take the stand which we now do. I have nothing but respect and admiration for our predecessors who have done so nobly and who hold so tenaciously to the old doctrine, but they cannot but see, if they will, that the reverence with which water power has always been mentioned is gradually disappearing.

CCCLXI.

COST OF LUBRICATING CAR JOURNALS.

BY L. S. RANDOLPH, BALTIMORE, MD.
(Member of the Society.)

IN studying the subject of the lubrication of car journals, the writer was led to the conclusion that the usual lubricating tests of coefficients of friction, durability, etc., were unreliable, as they did not pretend to reproduce the conditions of actual practice, giving neither the wear on the journal—which is more important than the coefficient of friction—nor the action when the journal was "flooded" with the lubricant, the condition which obtains in good practice.

In order to overcome these objections the following method was devised by the writer, which can perhaps be best explained by describing a test recently made, the official report of which is as follows:

Report of Test of K. Lubricant.

Test commenced July 18, 1888; test completed September 30, 1888.

No. of Journal or Brass.	Diameter of Journal or Weight of Brass. Before.	After.	Am't Wear.	Remarks.
1 Journal	$2\frac{31}{32}$	$2\frac{7}{8}$	$\frac{6}{64}$	With lubricant.
2 "	"	"	"	"
3 "	"	$2\frac{57}{64}$	$\frac{5}{64}$	Oil used as standard.
4 "	"	$2\frac{29}{32}$	$\frac{4}{64}$	" "
1 Brass	$97\frac{1}{2}$ oz.	72 oz.	$25\frac{1}{2}$ oz.	With lubricant.
2 "	$100\frac{1}{2}$ oz.	$71\frac{1}{2}$ oz.	29 oz.	"
3 "	$108\frac{1}{2}$ oz.	93 oz.	$15\frac{1}{2}$ oz.	Oil used as standard.
4 "	$101\frac{1}{2}$ oz.	84 oz.	$17\frac{1}{2}$ oz.	" "

REMARKS.

Amount of lubricant used, 15 lbs. Amount of oil used, $2\frac{1}{2}$ gals. $= 18\frac{1}{2}$ lbs.
Duration of test, 73 days. Mileage, 8,511 miles

Test was made on east truck of baggage car No. 12. Weight

on journal about 5,000 lbs., probably varied from 4,000 to 6,000 lbs.; maximum speed, 40 miles per hour.

Box packed with lubricant ran slightly warm all the time; box packed with oil seemed to be same temperature as the atmosphere.

The method of comparison is as follows:

X = relative value of the two lubricants, the standard being taken as unity.

a = value of brass per ounce wear.
b = " journal per $\frac{1}{64}$ ounce wear. } These figures are obtained by dividing the total value of the piece in place less the value of the scrap, by the amount in ounces, or sixty-fourths of an inch, necessary to be worn off before the piece is condemned.

$\mathfrak{z}$ = wear of brass in ounces.

z = wear of journal in sixty-fourths.

m = mileage.

$\mathfrak{z}'$ z' and m' = similar values for the standard.

We then have

$$X = \frac{(a\mathfrak{z}' + bz')\,m'}{(a\mathfrak{z} + bz)\,m}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

As a rule $m = m'$, but cases arise when that is not the case.

Applying the formula, we have first to determine the constants a and b.

The average weight of the brass is 104 oz.; this, at 1.2 cents per oz., = \$1.25. Allowable wear, 38 oz.; 104 − 38 = 66 oz. scrap at $\frac{1}{2}$ cent per oz. = 33 cents; hence net value of brass = \$1.25 − 0.33 = 92 cents; $\frac{92}{38}$ = 2.4 cents per oz. = a.

An axle weighs about 350 lbs.; at 2.4 cents = \$8.40 + labor = \$10.00; deduct 300 lbs. scrap at 1 cent = \$7.00. Three-quarters of an inch wear will throw an axle out of service; hence $b = \frac{7.00}{48}$ = 14.6 cents. Substituting in equation (1), we have

$$X = \frac{(a\mathfrak{z}' + bz')}{(a\mathfrak{z} + bz)} = \frac{2.4 \times 14.5 + 14.6 \times 4.5}{2.4 \times 27 + 14.6 \times 6} = \frac{105.3}{152.4} = 0.69. \quad . \qquad (2)$$

The oil used as a standard was an ordinary straight, reduced petroleum about 28° B. and 15° F. cold test, and cost about eight cents per gallon delivered in carload lots.

The value of the lubricant, taking into account the amount used, would be

$$X = \frac{18.2}{15}\, 0.69 = 0.83, \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

which with standard at 8 cents per gallon = 6.64 cts.

As a rule the amount of lubricant or oil used is not regarded, as there is so much waste, and so much is dependent on the condition of dust-guards, etc., that any attempt to regulate the amount consumed would vitiate the results to such an extent as to make them practically worthless.

When the coefficient of friction can be obtained with any degree of accuracy, it is well to take into account the relative amounts of coal consumed. This would give to equation (1) the following form:

C = value of coal consumed per mile per pound of train resistance.
W = weight on journal in pounds.
f' = coefficient of friction of standard.
f = " " lubricant.
d = diameter of journal in inches.
d' = " wheel "

The frictional resistance would be $= Wf \times \frac{d}{d'}$, and the value of the coal $= CWf \times \frac{d}{d'}$, and formula (1) becomes

$$X = \frac{\dfrac{(a\mathfrak{z}' + bz')}{m}\, CWf' \times \dfrac{d}{d}}{\dfrac{(a\mathfrak{z} + bz)}{m} + CWf\, \dfrac{d}{d'}}. \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Assuming a coefficient of friction for the standard of 0.01 and for the lubricant of 0.04, which are about the values they would have given, we can apply equation (4) to the test described.

To obtain the value of C, we have one pound of coal of about **14,000 B. T. U.**; calorific value would give 10,808,000 ft. lbs. of energy. This, with a boiler efficiency of 40%, and an engine efficiency of 10%, would give 432,320 ft. lbs., hence $\frac{5280}{432320} = 0.012$, lbs. coal per pound of resistance per mile.

With coal at \$1.00 per ton, C would be equal to 0.0006 cts.

Substituting these values in equation (4):

$$X=\frac{\left(\frac{a\mathfrak{z}'+bz'}{m'}\right)+CWf\frac{d}{d'}}{\left(\frac{a\mathfrak{z}+bz}{m}\right)+CWf\frac{d}{d'}}=$$

$$\frac{\frac{2.4\times16.5+14.6\times4.5}{8511}+.0005\times5000\times.01\times\frac{3}{30}}{\frac{2.4\times27+14.6\times6}{8511}+.0005\times5000\times.04\times\frac{3}{30}}\;.\quad(5)$$

$$X=\frac{.0123+.0025}{.0179+.01}=0.53.\quad\ldots\ldots\quad(6)$$

In formula 6 the first quantities in the numerator (0.0123) and in the denominator (0.0179) are from the wear of the journal and bearing; the others are due to coal consumed. It will be observed that the former, being much larger, have a much greater effect on the value of f than the latter, and consequently that the wear of journals and brasses is a much more important factor than coal consumption.

The value .04, assumed for the coefficient of friction of the lubricant, is somewhat high, but about correct, as it was undoubtedly a very poor lubricant.

DISCUSSION.

Prof. J. E. Denton.—Mr. Randolph communicated these results to me some time since, and I thought it was a great advance in the method of testing oil to put it on a car and keep account of what occurred in the interval of time long enough to make an appreciable wear. I believe that most of the railroad laboratories have long since decided that the greatest element of cost in lubrication is the metallic wear which takes place on the axle and the brass. I understand that the lubricant of which he speaks was what is called a cooling compound, and I want to speak of an interesting deduction from that fact. The cooling compounds which are sold are a

grease of some kind mixed up with talc or sulphur, or in this case bituminous coal. A sample of this which was sent to me was made up of bituminous coal mixed with grease, and yet it ranks as a cooling compound for hot boxes, and you notice that it wears the journal considerably more than the oil. The theory of the cooling compounds is simply this, that if the journal is about to become hot, and if you can catch it before it is abraded sufficiently to get hot right away, that is in a few seconds—if you could open the box and get on a piece of sandpaper, you could smooth it, and it would cease to get hot. Now, the theory is that the talc or bituminous coal, if you can get it on the journal in time, will smooth the bearings. In experiments made personally on oil testers, I have noticed the friction in gearing in time to stop the machine and discover the spot where the surfaces were roughening; I have rubbed the rough surface smooth with an oil-stone and put the brasses back, and the journal has gone on again all right. A cooling compound is therefore a good thing for a hot box, but if you use it for a regular lubricant it necessarily makes more wear than oil. There is so much mathematics in the paper that it is a little beyond me to read it here in a minute, but I believe it comes out something like this: that metallic wear costs a cent, the cost of coal to overcome the friction is about one-third of a cent, and the cost of the oil, at 15 cents per gallon, one-sixth of a cent; the metallic wear is by far the greatest factor. I suggested this method to two other roads who wanted some other way of testing oil, rather than testing it in the laboratory. They tried it, but they met this difficulty, that the greatest wear was endwise on the brasses. The collar of the journal cut in, and that prevented using it as an every day test. Mr. Randolph, I think, avoids that, because he has a very light service, but it establishes the fact that the great element is metallic wear, the second the coal, and the last the oil. I believe Dr. Dudley, of Altoona, was the first to discover this fact.

Mr. C. J. H. Woodbury.—The actual determination of the coefficient of friction of lubrication by direct measurement is, in the nature of things, a laboratory test requiring a great deal of delicacy. The differences in physical condition of pressure, temperature, velocity, and the relative abundance of lubricant all modify the results, and even when these results are accu-

rately obtained they pertain merely to the conditions of the test, and not to the exact conditions under which the lubricant is used.

Several years ago I made a number of examinations of the frictions of various journals by the use of two thermometers, a method which I adopted at the suggestion of Mr. Edward Atkinson, and referred to in a paper* on the subject of "Measurements of Friction of Lubricating Oils," which I read before this society in November, 1880. The apparatus is very simple. A hole in the cap of the journal contained a very thin copper tube, closed at the lower end and reaching to the shaft. A

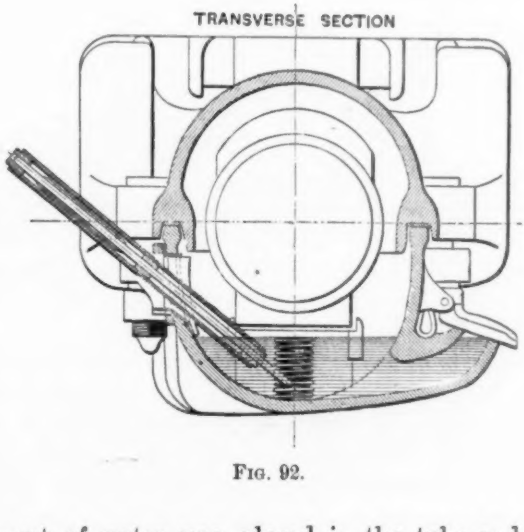

FIG. 92.

small amount of water was placed in the tube and the thermometer bulb inserted therein. A second thermometer hanging in the air near to the journal gave the data for learning the increase of temperature due to the friction of the journal, and from this was deduced the relative friction of various lubricants and different methods of lubrication.

At the Paris Exposition, in the exhibit of the Compagnie des Chemins de Fer de l'Est, there was an apparatus showing an identical method of judging various lubricants used on the car axles; one thermometer reaching to the cap of the journal (Figs. 92 and 93), and the other to the oil beneath.

* Trans. A. S. M. E., Vol. I., page 78, No. VII.

Mr. T. R. Almond.—In considering this subject it seems to me very necessary that the exact condition of the bearing should be thoroughly well known. I have had several occasions to notice what quite surprised me. I make a device for transmitting motion at right angles, and some of the machines after they are put up and started will get so hot that the oil will boil inside of them, and the question has been what was the cause of it. That has been something very difficult to

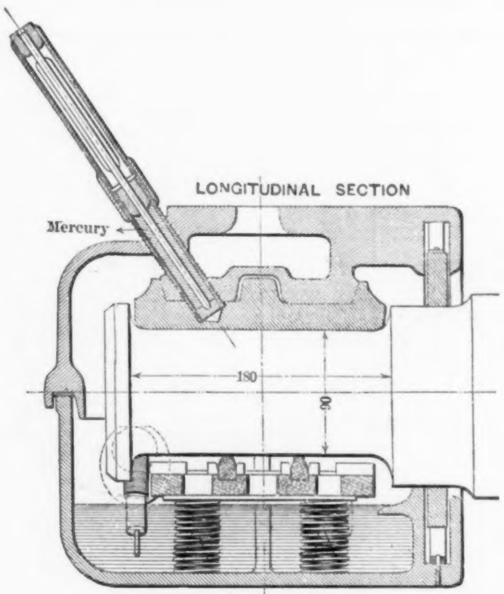

FIG. 93

find out, but I finally discovered that the reason why the oil got hot was because the bearings would bear slightly to one side, as it were, and create a sort of wedging tendency hardly perceptible; but an amount of scraping in the bearings equal to one or two thousandths of an inch would be sufficient to bring the machine down to the condition where there would be no further heating. I have it very well fixed in my mind now, that in considering any subject of this kind we should pay very careful attention to the character of the surfaces in contact. The quality of the lubricant cannot be

determined until we thoroughly know that the surfaces in running contact are perfect.

Mr. W. S. Rogers.—In this test on that car I think the alignment was all right. I have had some experience with hot boxes, and I can bear out what Professor Denton says in regard to sandpaper. It is customary on a good many Northwestern roads, among trainmen, when they get a hot box and it bothers them considerably, to take out all the lubricant and packing and fill up the box with fine sand and run it about fifteen miles, and then let it cool. Then they pack it with good, nice tallow,—if they cannot get that somebody will go and buy tallow candles at a country grocery,—and it doesn't bother them any more. But I found, in my experience on the Baltimore & Ohio Road, where we had a great many hot boxes, and a very good lubricant, that we had very poor brasses, and our own solution, when we got into trouble with our hot boxes, was to steal brasses from the Lake Shore Road. (Of course the officials were not cognizant of our little scheme.) I think the quality of material in the "brass" itself has equally as much to do with a thorough test as the lubricant.

Mr. John Dick.—I am familiar with an experiment tried with pure copper bearings. Two boxes were made from the regular patterns and put under the tender of a heavy freight engine, and also two new boxes of the regular brass mixture. All were carefully weighed before and after the test. These were run six hundred miles, and at the end of the test the copper bearings were found to have worn less than the others, the journals were in better condition, and these journals were oiled only once in two hundred miles, giving no trouble from heating. The other journals required to be oiled frequently, and heated more or less all the time, which latter experience, I believe, is common to all railroads, particularly on tender bearings.

*Mr. L. S. Randolph.**—In the experiments mentioned there was very little end-wear of the brasses, or not enough to be measured accurately. Where there is much it is measured and the calculated amount allowed for. Usually this can be done within the limit of error of the ordinary scales.

There is another point in the theory of the action of cooling compounds, which is that the solid ingredient fills up the rough places. Plumbago seems to act in this way as well as

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* Author's Closure, under the Rules.

soapstone, although the latter often seems to act as Mr. Denton states.

It has been the ordinary custom always to run duplicate tests as in this case, the average being taken. Where there is a great discrepancy between the results gained, an investigation always shows some marked difference in the conditions.

CCLXII.

PHILOSOPHY OF THE MULTI-CYLINDER, OR COMPOUND, ENGINE; ITS THEORY AND ITS LIMITATIONS.

BY R. H. THURSTON, ITHACA, N. Y.

(Member of the Society and Past President.)

WERE it possible to construct a steam-engine of which the theory should be a thermodynamic one, an engine in which the only waste of energy should be that known as the necessary thermodynamic loss, its theory would be most simple and most satisfactory. The efficiency of the engine and the quantities of heat, steam, and fuel demanded for its operation at a given power, would be simple functions of the physical properties of the steam and of its ratio of expansion. The engineer, in constructing its theory, would only concern himself with the quantity of heat imported into the machine, the temperatures of the initial and terminal portions of the expansion line, and the relation of initial to back pressures. As was well stated by Rankine, a generation ago, nearly, the essential facts are the magnitudes of the pressures and volumes of the steam and the extent of adiabatic expansion, and it matters not whether the engine be one of a single cylinder or a multi-cylinder engine of indefinitely extended complexity. For this, the ideal case, as the writer has been accustomed to call it, the indicator diagram represents precisely the amount of transformation of heat-energy into mechanical work, and the ratio of its measure in units of work to the mechanical equivalent of the total quantity of heat-energy supplied to the engine, while doing that work, is the measure of the efficiency of the engine, as it is of the thermodynamic efficiency of the working fluid. The thermodynamic efficiency, the dynamic efficiency of the machine, and the total efficiency of the engine are here identical.

To ascertain how much heat, steam, and fuel are demanded by such an engine for the performance of work, it is only necessary to measure the quantity of work done by the steam upon the pis-

ton, as shown by the indicator, and to divide this quantity by the energy received by the engine from the boiler; the quotient is the efficiency of the engine. As the operation of the engine approaches more nearly the conditions of best effect, the magnitude of this measure of efficiency approaches a limit which is expressed by the quotient of the range of temperature worked through to the absolute temperature of the working fluid at entrance into the engine. Were all heat received completely utilized, and the full thermodynamic equivalent realized in work performed, the quantity of steam demanded per hour and per horse-power, under the conditions now common in practice, as to initial and final temperatures, the engine being assumed perfect, structurally, would be about two and a half pounds, for efficiency unity, or not far from ten pounds for the best real cases, no wastes occurring. The excess of the actual consumption of fuel, in the best engines, above this last figure measures the sum of all wastes in real engines due to imperfections other than of thermodynamic cycle. Thus, the best work of the compound mill-engine may be taken as about sixteen pounds of steam per horse-power and per hour, where the thermodynamic efficiency is, as just assumed, about twenty-five per cent. The wastes amount, in this case, therefore, to about six pounds per horse-power and per hour, or sixty per cent. of the ideal consumption. This comparison is easily made by a method first presented in full by Rankine, a generation ago, and which enables the thermodynamic efficiency to be easily computed for any given case.* Examples will be presently given of its application.

THE WASTES OF THE STEAM-ENGINE comprehend two principal classes: the external and the internal wastes; and these latter are of two distinct kinds. We may classify them thus:—

(1.) External wastes; consisting of those losses of heat without transformation, which are produced by the conductivity and the radiating power of the materials of which the heated parts of the engine are composed. Heat which might otherwise be utilized by conversion into mechanical energy, is conducted or radiated to the adjacent parts of the engine and to surrounding objects.

This form of waste is of small amount, comparatively, and can readily be kept down to an unimportant proportion of the heat supplied from the boiler by properly covering exposed parts of the

* The Steam-Engine and other Prime Movers, pp. 383–412.

engine with non-conducting covering. Five per cent. should probably represent as large a percentage as is to be reasonably expected in good practice.

(2.) Internal Wastes; consisting of two parts:

(*a*.) Thermodynamic, unavoidable, losses of heat rejected at the lower limit of temperature of working fluid;

(*b*.) Wastes by internal conduction and storage of heat, followed by later rejection with the exhaust steam.

To these are to be added:

(3.) Wastes of mechanical energy.

Of the losses, the first, (*a*), represented by the fraction, as a minimum, $\frac{T_2}{T_1}$ of the heat supplied, is, for any given set of initial and final temperatures of working fluid, a fixed and irreducible quantity, and one which measures the defect of efficiency of the perfect engine working between the given temperatures. The second, (*b*), is a quantity of variable amount, capable of amelioration by one or all of several known expedients, and reducible from the enormous proportion observed in small and ill-designed or badly constructed engines to a very moderate amount in large engines of good type. The last item, (3), is one which is seldom large in good constructions, as compared with the magnitudes, at least, of the other kinds, and may in some cases, by careful design, good construction and skilful management, be brought down to less than five per cent. in non-condensing, and to perhaps ten per cent. of the total energy in condensing, engines of simple forms and high mean working pressures. The unavoidable thermodynamic waste is rarely less than seventy-five or eighty per cent., and the internal wastes by conduction and storage with subsequent rejection, by cylinder or internal condensation, as it is customarily called, and by leakage, range from ten per cent., as a minimum, perhaps, to twenty-five or thirty per cent., in good engines, to fifty per cent. in many cases, and even to much more than the latter proportion in exceptional cases. It is this which constitutes, ordinarily, the great source of loss and inefficiency of the real, as distinguished from the ideal, engine. Leakage in well-built engines may be neglected as unimportant; but internal condensation is usually both serious in amount and extremely difficult to check effectively.

Since it is easy to prevent serious losses by external transfers of heat, by leakage, or by friction of engine, and since, as is well understood, the thermodynamic waste is unavoidable, and for any

given case unalterable by the engineer, it is obvious that the direction in which he must look in his endeavor to further improve the economical performance of the engine, is that which leads toward the reduction of internal wastes by cylinder condensation.

The Method of Internal Waste by Condensation, as is now well and generally known, and as is now shown by all authorities,* consists in the absorption and storage of heat in the metal constituting the internal surfaces of the cylinder, at the commencement of the stroke, and to an extent which is determined by the difference of temperature at the moment existing between the prime steam and those surfaces which have been exposed, during the exhaust stroke, to the cooling influence of the comparatively cold steam passing into the condenser or into the atmosphere. This stored heat is, later, rejected during the terminal portion of the expansion period, and during the succeeding exhaust stroke, to the mass of steam exhausted, in its turn, at the opening of the exhaust passages. The quantity so wasted varies with the weight of steam worked thermodynamically each stroke, with the area of surface exposed to this action, with the period of exposure, and with the range of temperature worked through during the cycle. It is thus always an increasing proportion when the ratio of expansion is increased, and as the size of engine doing a given amount of work is increased; it is diminished by increasing engine-speeds and by any expedient which reduces the storing capacity of the interior of the cylinder. Since it is always an increasing function of the ratio of expansion, there always will be found, in any engine and under any given conditions of operation in other respects, a point beyond which further increase in the magnitude of that ratio will result in a loss by cylinder condensation greater in amount than the gain due to extended expansion, a circumstance which has been now known, for many years, as placing an early limit to the expansion of steam, and to the gain to be anticipated as a consequence.†

The method of variation of this waste was qualitatively determined by Clark about 1850; was closely gauged by him, both as to magnitude and as to its effect in limiting the ratio of expansion; was quantitatively investigated by Hirn and by Isherwood afterward, and was finally made the subject of an investigation, under

* See especially the works of Clark, Hirn, Isherwood, and Cotterill, and more recently of Dwelshauvers-Dery.

† Trans. Am. Soc. Mech. Engrs., 1882, *et seq.*; Trans. N. Y. Acad. Sci., 1882; Brit. Assoc. for Adv. Sci., 1884; Journ. Franklin Inst., 1882, *et seq.*

the supervision of the writer, by Messrs. Gately and Kletsch, on a plan schemed out by the writer some years earlier,* in which it was endeavored to ascertain with some degree of accuracy the method of variation of the waste with variation of each of the essential conditions affecting and determining it. The result of this research in brief was to show that the waste varied, in the cases studied, sensibly as the square root of the ratio of expansion, and as the time of exposure, and was subject to a very slow decrease as the pressures adopted increased, the engine being worked condensing; decreasing about twice as rapidly, the condenser being thrown off.†

Variation with ratio of expansion was also capable of being expressed with great accuracy by an hyperbolic expression, the product of areas of surface exposed up to the point of cut-off and the percentage of condensation being found sensibly constant. Under ordinary working conditions, the steam pressure being about sixty pounds per square inch, by gauge, the cut-off at one-third, and the speed of piston 554 feet per minute, and of rotation sixty-eight revolutions, the condensation was about one-third, or the equivalent of fifty per cent. of the total consumption of a similar engine having a non-conducting cylinder, and thus free from this waste. Reduced to quantity of steam and of heat wasted, per square foot of surface exposed to point of cut-off, per minute of exposure, and per degree of range of temperature between prime and exhaust steam, Professor Marks finds the co-efficient to be 0.02 pounds, or 18 B.T.U., nearly, a result closely confirmed by the investigations of the same author, taking Hill's experiments for comparison, and also corroborated by the later work of other investigators.

These facts and laws being established, it becomes possible to determine the behavior of steam entering any given cylinder, and its method of working and of waste in any engine. Common experience, as well as theoretical considerations based upon the investigations already made, prove that it is impossible to expand steam in the ordinary single-cylinder engine with satisfactory gain of efficiency beyond a point variable with the conditions assumed, but which may be roughly taken as not far from that giving a ratio of expansion equal to about one-half the square root of the steam

* Jour. Frank. Inst., Oct. and Nov., 1885. Trans. Am. Assoc. for Adv. Sci. Ann Arbor Meeting, 1885.

† The proportion thus wasted in these experiments were from $0.18\sqrt{r}$ to $0.197\sqrt{r}$. The writer takes, for similar cases, $c = 0.2\sqrt{r}$ in later work.

pressure, measured from vacuum for the condensing engine of common type, or as measured by gauge for the non-condensing engine. The waste by internal condensation increasing as the point of cut-off is shortened up, the loss, after a time, compensates the gain by increased expansion, and a point of maximum economy is passed at a very early stage for the older types of engine and later, but still at a comparatively low value of the ratio of expansion, for modern engines. For example, the old beam engine, such as was used in American rivers, or in the coast trade, a generation ago (1850), with steam at twenty-five pounds by gauge and a low piston speed, had a ratio of initial to back pressure plus friction of about 8 to 1; but its best ratio of expansion for efficiency of fluid was about 2 or $2\frac{1}{2}$. The same type of engine, later, with twice the pressure per gauge, had values of these two ratios of about 16 and 4 or 5 respectively. The ratio of expansion for maximum efficiency of working fluid was thus but about one-fourth that which thermodynamic theory, unqualified, would dictate. As engines have been improved this discrepancy has been reduced, but it still remains, with the best of engines, considerable.

The amelioration of Wastes thus becomes an important matter. It thus happens that the efficiency and economy of operation of the single cylinder, the "simple" engine, is at all times limited by this very serious internal waste, and the question which all engineers since Watt have been endeavoring to solve, is: In what manner may we best proceed to eliminate or ameliorate this loss? The three methods which have been found advantageous, and, in special cases, fairly effective, are:

(1.) Superheating;
(2.) Steam jacketing;
(3.) "Compounding."

It is evident that, if the steam can be introduced into the engine at such a temperature that the cooling action of the metal of the cylinder will not cause its condensation initially, and the stroke may be performed without condensation in consequence of doing work, no loss of heat from the cylinder can take place by re-evaporation; and if no such loss occurs, the waste of heat at entrance, in turn, by initial cooling, will be reduced. Superheated steam, also, is a non-conductor and a non-absorbent of heat, precisely like the permanent gases. It is thus, also, less liable to this waste. But it is found in practice that superheating beyond a very moderate degree, perhaps 100 degrees Fahrenheit, is inadvisable on account of risks

of injury to engines and cost of repairs to superheater, which more than compensate its advantages. It has come to be regarded as an auxiliary in economizing, not as a remedy for interior wastes.

Steam-jacketing is another and a common partial remedy for this waste. By surrounding the steam-cylinder with the steam-jacket, it is possible to produce, in part, the effect of superheating; that is, to secure dryer steam in the engine throughout the stroke. The amount of re-evaporation, during the period succeeding cut-off and up to the closure of the exhaust-valve, and the quantity of heat of which the cylinder is thus robbed, measures the amount of initial condensation and waste and the weight of steam which must be supplied in excess of the thermodynamic demand to compensate that loss. The effect of the addition of a steam-jacket depends upon the conditions of operation of the engine, largely, and it may be productive of marked advantage, or, under unfavorable conditions, of no important useful effect. With steam initially dry or superheated, the jacket is probably always decidedly helpful; but with wet steam it is of comparatively little value, even if not sometimes a positively wasteful adjunct. High-speed engines derive less advantage from its application than slow-moving machines; and compound, or multi-cylinder engines, are less dependent upon it for economy than are simple engines. The saving effected in ordinary cases by its use may be taken as averaging ordinarily in mill engines about 20 per cent.; and about the same gain is attained by effective superheating within the usually practicable range. The two devices in conjunction may be expected, perhaps, to give a gain of something like 30 per cent. as compared with the standard forms of unjacketed simple engine working with slightly wet steam. The addition of either expedient to the latter practice, if properly performed, considerably increases the magnitude of the ratio of expansion at maximum efficiency of fluid. Where it would ordinarily be approximately equal to one-half the square root of the pressure, as above, it might become, with superheating or with steam jacketing, a figure as much as 30 or 40 per cent. higher; and both expedients together might nearly double the profitable ratio of expansion. The assumption is commonly made that the superheating is retained throughout the stroke and that steam-jacketing may be relied upon to keep the working charge dry and saturated throughout the stroke; but neither of these hypotheses, as employed in the theory of the engine, is practically correct.

"*Compounding*," or the use of the multi-cylinder engine, in which the steam exhausted from one cylinder is again worked in a succeeding one, is the most familiar of devices for extending the economical range of expansion and increasing the efficiency of the engine. The limit to the useful extension of the expansion of steam in a single cylinder is found to be determined by the magnitude of the wastes incurred in the operation of an engine of which the working cylinder is a good conducting material. Any method of reducing this waste of heat internally will enable the efficiency of the engine to be increased by further profitable extension of the ratio of expansion. Common experience with the best constructions, and considerations which need not be here reviewed, show that the engineer may reasonably expect, by good design, construction and management, to secure an economy of steam which is fairly measured by the following table, the ratios of expansion, r, taken being, for each case, those which give best results for a given engine: *

STEAM PER HORSE-POWER PER HOUR,

at best ratios of expansion in best engines.

r	3	4	5	6	7	8	10	12	15	20	25	50	75
lbs.	32	27	25	22	20	20	19	17	16	15	15	11	9
kgs.	15	12	11	11	9	9	9	8	7	7	7	5	4

and ten per cent. better figures than some of these have been actually reported in peculiarly favorable cases.

Assuming it to be possible to divide the waste due to cylinder condensation and leakage by two or more, it is evident that the limit to economical expansion and transformation of heat into work will be set correspondingly further away. This is precisely what is done by the multi-cylinder engine. The internal wastes are reduced approximately to those of a single cylinder, and the gross percentage of waste is made less in the proportion of this division. The heat and steam rejected as waste by internal transfer without transformation from the first cylinder, is utilized in the second nearly as effectively as if it were received directly from a boiler at the pressure of rejection from the first cylinder. In so much, therefore, as the pressure can be increased and the increase utilized by the addition of another cylinder, gain is secured. If the total

* Several Efficiencies of the Steam Engine ; Trans. A.S.M.E., and Jour. F. Inst., 1882.

ratio of expansion can thus be raised, under the best working conditions for each case, from, we will say, four up to eight, we should hope to secure a reduction of coal consumed from two and a half, we will say, to two pounds per horse-power and per hour, which is about the average figure in good practice.

The practical questions thus meet the engineer: To what extent can this principle be availed of? What range of pressure and what ratio of expansion should be assigned to a single cylinder? and how many cylinders should be adopted to give best results with the highest steam pressure practicable for a specified case? Common experience aids in solving this problem by showing that the very best results are ordinarily obtained, in each class of multi-cylinder engine, when, the engine being properly designed for its work, terminal pressure for the system can be economically made something above the sum of back pressure in the low pressure cylinder, plus friction of engine. This total may be usually taken, probably, at about eight or ten pounds above a vacuum. The latter figure will be here assumed.

The Fundamental Principles are now easily perceived. There are three main facts upon which to base our theory of the multi-cylinder engine. These are:

(1.) *Economical expansion in a single cylinder has a limit, due to increasing internal wastes, which is found at a comparatively low ratio of expansion.*

(2.) *The method of expansion may be, for practical purposes, such as are here in view, taken to be approximately hyperbolic; the terminal pressure being something above that which corresponds to the sum of all useless resistances, and which may be here taken, as, for example, about ten pounds per square inch above a vacuum. The division of the initial pressure by this terminal pressure will thus give an approximate measure of the* desirable *ratio of total expansion for the best existing engines.*

(3.) *All steam entering any one cylinder will be rejected,* as steam,* *into the succeeding cylinder, external wastes being neglected, and into the condenser; and the full amount of steam condensed at entrance by absorption of heat by the interior surfaces of the cylinder will be re-evaporated later, and will pass into the condenser or into the next cylinder, and heat transferred in the one*

* This the writer would denominate Hirn's principle. See a paper by Dwelshauvers-Dery in the *Bulletin de la Société Industrielle de Mulhouse,* Oct., 1888, on the theory of single-cylinder engine.

direction, in the one process, will be transferred in precisely equal amount in the opposite direction in the other.

This last point is a very important one, and is very easily established. The cylinder, when in steady operation, is neither permanently heated or permanently cooled; no progressive heating can go on, as it would, in that case, become heated above the temperature of the steam and become a superheater; no progressive cooling can occur, since, in that case, the cylinder would become a condenser of indefinite capacity. It must, therefore, transfer to the next element of the system all the heat which it receives, assuming that external radiation and conduction may be neglected and that the Rankine and Clausius phenomenon of internal condensation, by transformation of heat into work, is ignored. It also further follows that the introduction of one or of many cylinders between the terminal element and the boiler does not, through cylinder condensation, affect the operation of the latter cylinder, however great that condensation may be, provided the operation of the added elements is effected by raising the steam pressure commensurately, leaving the final element of the series the same initial pressure as before. The total waste by this form of loss is thus evidently measured, in the case of the multi-cylinder engine, by the maximum waste in any one cylinder. If all are equally subject to this loss, the rejected steam of re-evaporation from any one cylinder, as the high-pressure cylinder, supplies precisely what is needed to meet the waste by initial condensation in the next; and so on through the series. Thus the use of a series of cylinders, in this manner, divides the total waste for a single cylinder, approximately, at least, by the number of cylinders; and it is in this manner that the compound system gives its remarkable increase of efficiency. As stated by the writer, many years ago, "The serious losses arising from condensation and re-evaporation within the cylinder, and which place an early limit to the benefit derivable from expansion, affect both types of engine, and so far as seems now known, equally;" * but the modern type permits the interception of the heat wasted from one cylinder, for utilization by its successor, in such manner that the total waste becomes, practically, that of the low-pressure cylinder alone. If any one cylinder wastes more than another, the total waste is, as above stated, measured more nearly by the loss in the most wasteful member of the system.

Thus the three principles which have been above enunciated

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* Vienna Report, 1873.

give a means of constructing a philosophy of the multi-cylinder engine, which will meet the essential needs of the designer and of the student of its theory. The first principle shows that a limit existing to economical expansion in a single cylinder, the advisable number of cylinders in series may probably be determined, when that limit is ascertained, either by experiment, by general experience, or by rational theory and computation. The second principle shows that we may find a tentative measure, at least, of the desirable total ratio of expansion for maximum density, when the best terminal pressure for the chosen type of engine is settled upon. This total range is divided by the admissible range for a single cylinder, or, perhaps better stated, the total ratio is a quantity which should approximately equal the admissible ratio for a single cylinder, raised to a power denoted by the number of cylinders. Combining thus the two considerations referred to, we obtain a determination, probably fairly approximate, of the proper number of cylinders in series. The third principle permits an estimate to be made of the probable internal wastes of the series, and the probable total expenditure of heat and of steam, and a solution of all problems of efficiency for the compound engine, of whatever type.

The first step in the process is evidently the determination of the best ratio of expansion, under the assumed conditions of operation and for the given type of engine, for a single cylinder; then the best ratio of expansion for the series, all things considered, this study being made from the financial standpoint, as must be every problem which the engineer is called upon to solve. It is not the thermodynamic, nor the fluid, nor even the engine efficiency, which must be finally allowed to fix the best ratio of expansion; but it must be the ratio of expansion at maximum commercial efficiency, that which will make the cost of operation at the desired power a minimum for the life of the system.* The total ratio being settled upon, and that allowable, as a maximum, for the single cylinder, it is at once easy to determine the best number of cylinders in series. The first-mentioned ratio is that at maximum commercial efficiency, as just stated; but the second must be taken as that which gives the highest efficiency of engine, the back-pressure in that cylinder, and the friction of the cylinder taken singly, being considered, together with its proper proportion of the friction of the engine as a whole.

* See papers by the writer on the efficiencies of engines, as per references already given.

Studying the method of distributions of waste among the several cylinders of the multi-cylinder engine, it will be observed that, since the pressures increase more rapidly than the temperatures, the range of temperature in the high-pressure cylinder is greatest; while, the same weight of steam passing through the whole series, the low-pressure cylinder presents the largest area of condensing surface in proportion to steam used. These differences are to a certain extent, though not wholly, compensatory. It may be assumed, however, without serious error, that the necessity of applying jackets or other methods of reducing internal wastes, will apply substantially as imperatively to one cylinder as to another, and that the adoption of a common ratio of expansion for both or all cylinders, or of apportioning the ratios with reference to the equal division of power among them, will be found perfectly admissible and will introduce no serious avoidable loss. Authorities have greatly differed in their views as to the relative advantage of jacketing one or another cylinder; but it is at least safe to jacket all, and probably, as above indicated, best to do so. The importance of the jacket evidently becomes less as other expedients for reducing wastes of this kind are adopted and are made more effective; as by increasing speed of engine, by superheating, by reheating between cylinders; and cases may be imagined in which the jackets may cease to have sufficient value to justify the acceptance of the risks and expense incurred in their employment. The same is true of the more complicated forms of valve-gear needed to secure an approximation to the ideal distribution of steam.

The extent to which expansion may be economically carried in a single cylinder will vary somewhat with the initial temperature and pressure, and with the physical condition of the working fluid; but it may be taken as ordinarily not less than two-and-a-half expansions for unjacketed engines with wet steam and three or four for the better class of engines. The total expansion ratio thus becomes, for several types of multi-cylinder engines, as below:

MULTI-CYLINDER ENGINES.

No. cyls.	1	2	3	4
r	2.5 to 3	6.25 to 9	16 to 27	40 to 81
p_1	25 to 30 lbs.	60 to 100 lbs.	120 to 300 lbs.	350 to 800 lbs.

Expansion is here assumed to be approximately hyperbolic, and the terminal pressure to be eight or ten pounds per square inch.

General experience to date thus indicates that a triple expansion engine should do best work up to a pressure of about 250 or 300 pounds, and that the four-cylinder engine should be adopted from that point up to the highest pressures likely to be adopted in the steam-engine, the double expansion compound serving its purpose well below the lowest figures above assigned to the triple engine. Any of the four types of engine may be made to overlap the range assigned its labor by suitably providing against wastes occurring within the engine by increased speed, by superheating, by expedients giving higher effectiveness to the jackets, or other methods of improvement. Any system which increases the efficiency of the simple engine will improve the efficiency of the compound, and will correspondingly increase the range of pressure through which it will give satisfactory gain as compared with the former.

The influence of the several economical expedients recognized as useful in other forms of engine, as superheating, jacketing, and high speed of engine, may readily be perceived when the method of operation of the multi-cylinder engine is understood in its relations to heat-transfer and heat-transformation. We may consider them in their order.

SUPERHEATING the steam transferred from boiler to engine results in the supply of a fluid which may surrender a certain portion of heat, measured by the product of its specific heat as a gas into the range of superheating and into its weight, to the metal of the working cylinder without the production of initial condensation. If this quantity is equal to or greater than the loss of heat during expansion and exhaust, there will be no initial condensation, and the waste from the high-pressure cylinder will be nearly that due to the passage of a gas through it under similar conditions of temperature and expansion, a comparatively small quantity, since any substance in the gaseous state possesses low conductivity and slight power of absorption and storage of heat. Should the superheating be in excess of this amount, the steam will not begin to condense until a later period, perhaps not at all, the only demand being now for heat to supply the amount required to keep the steam dry and saturated while expanding and doing work. If the superheating be less than the first mentioned quantity, initial condensation will be reduced but not entirely prevented. It is probably never the fact, in practice, that it is possible to secure, safely and economically, so much superheating as is needed to keep the steam dry

throughout the stroke.* In any case, the quantity of heat represented by the superheating will be a gauge of the amelioration of wastes by internal transfer of heat in every cylinder of the series. The steam leaving the high-pressure cylinder will be to that extent dryer than it would otherwise be; and this will be true of the succeeding cylinder or cylinders.

Were there no other disappearance of heat than that due to cylinder condensation, superheating at the first of the series would give superheating at each of the others. In so far as condensation doing work, such as was pointed out by Rankine and Clausius, takes effect, and so far as other wastes by transfer without transformation occur, to that extent will the gain, as observed in successive passages from cylinder to cylinder, be reduced, though the improvement of the working conditions above asserted will be none the less real. Each cylinder will have wetter steam than the preceding, in proportion as the condensation doing work and the losses by conduction and radiation increase, as a total, cylinder by cylinder. Superheating at the high-pressure cylinder will produce a favorable effect all through the series, including the low-pressure cylinder. Cylinder condensation will, nevertheless, cumulatively increase through the series, in consequence of the fact that the wetter the steam entering any one cylinder the more the condensation and the wetter that leaving it, both by this initial increase of humidity and by the additional moisture coming from the Rankine and Clausius phenomenon, and from the loss by transfer to surrounding bodies. This action will, however, be the less observable and the le-s important in its effect as the moisture of the entering steam and the magnitude of the waste by initial condensation become greater. The more nearly the total proportion of water in the mixture approaches one-half, the more nearly does this phenomenon become a vanishing quantity. It may probably be neglected entirely in the computation of efficiencies for a large proportion of the engines in use, without introducing sensible error, and very probably may be neglected in all cases without invalidating conclusions reached, ignoring it. On the other hand, superheating is not likely ever to produce much effect upon this action. Could we superheat safely and satisfactorily to the extent of doubling the absolute temperature of the steam at entrance into the engine, we might have a

* In one case reported to the writer an initial superheating of 500° F. was required to give 50° F. superheating at exhaust; 100° F. has usually been considered a practical maximum superheat.

"superheated steam-engine," but this is not yet practicable, and, until it becomes so, it is not likely that the best engines will entirely satisfy our theory in this respect.

STEAM JACKETING, the expedient devised by James Watt for the very purpose of reducing wastes by internal condensation, a phenomenon of which he was the discoverer, is a method of approximately "keeping the cylinder as hot as the steam which enters it," as Watt put it, in order that no such chilling of the entering steam may occur. We are interested in the answer to the question: To what extent and in what manner is the jacket advantageous in the compound or multi-cylinder engine? Authorities disagree, even where they have themselves had large practical experience. It is sometimes advised to jacket only the high-pressure cylinder; sometimes to jacket only the low-pressure cylinder, and sometimes to jacket the whole series, whether one, two, or three or more. The philosophy of the multi-cylinder engine, as above outlined, would obviously indicate that, to secure maximum good effect, assuming the jacket on the whole desirable at all, the best system is the latter, and that, since the waste of the engine is measured by the waste of its most wasteful member, to omit the jacket from any one cylinder insures that the aggregate loss of heat in the whole engine will be increased by just the amount by which waste is increased in that one cylinder by such omission.

The resulting effect, in detail, is evidently the following: Assume the intermediate cylinder to be unjacketed. That cylinder, being exposed to a wider range of heating and cooling action as it alternately takes steam and exhausts it, is subject to a greater waste by internal condensation than either of the others; it thus discharges into the next cylinder an equal quantity of heat and steam, but it does less work than it would have otherwise done, and to that extent produces decreased efficiency. Assume the high-pressure cylinder unjacketed, it demands more steam from the boiler, as it condenses a larger proportion of that entering by this process of initial liquefaction; it is thus itself more wasteful, and, furthermore, transmits to the succeeding cylinders a larger quantity, and therefore a more uneconomical apportionment, of steam than it would otherwise have released. In proportion as its own efficiency is thus reduced, it reduces the economical working of the whole; and, in proportion as the steam rejected from it is a less economical storehouse of heat for use in the other cylinders, they are in turn rendered less efficient. The low-pressure cylinder

being left unjacketed, it becomes more wasteful in proportion to the increased initial condensation thus permitted, and the whole system is again, to that extent, given impaired efficiency. In neither case, however, is the efficiency of the engine, as a whole, impaired nearly as seriously as in the case of the simple engine. The increased loss is mainly confined to the single cylinder left unjacketed. It is readily seen, however, that, to secure maximum efficiency, it is as essential to jacket the cylinders of the compounded engine as that of the simple engine. The question which actually arises in practice, for the designing engineer, is whether it will *pay* to jacket at all or not. It can at once be seen that it is not as important, in a financial sense, that the multi-cylinder engine be jacketed as it is to jacket a simple engine of similar range of expansion. The value of the waste due to omission of the jacket is less as the number of cylinders is the greater. It is also seen that those conditions which may make it undesirable, as a matter of finance, to jacket the simple cylinder, make it still less desirable in the compound or multi-cylinder engine. As piston speeds are increased, for example, the necessity of the jacket decreases and the limit at which it will pay to dispense with it is sooner reached in the multi-cylinder than in the single cylinder engine. It is this principle which justifies the now not uncommon practice of omitting jackets from marine engines which are driven up to 1,000 feet a minute; while pumping engines, in which the speed is always very low, must always be jacketed if high duty is demanded.

HIGH ENGINE SPEED, the most modern device for reducing internal wastes, as well as of decreasing costs of engine construction and weights of machine, is evidently a matter of less serious importance as the number of cylinders is increased; yet it is equally evident that, to secure maximum efficiency, it is essential that the time of exposure to the action of the wasteful influences in any one cylinder be made a minimum. At modern and customary speeds of piston and of rotation, the value of the other expedients for improving performance is much less than formerly; but all are to be adopted where it is hoped to secure such high efficiency as is coming to be demanded of the designing and constructing engineer. The advantage of further progress in this direction, now that piston speeds of $V = 500 \sqrt[3]{S}$ and a velocity of rotation equal to $R = 250 S^{\frac{2}{3}}$, and upward are becoming usual, does not seem likely to be great; and, except where superheated or thoroughly dry steam can be absolutely insured at all times, the risks attending increased

speeds seem also likely to retard this advance. So long as the advantages of further gain in this direction are safely attainable for the simple engine, they are still desirable and attainable in the multi-cylinder machine.

NON-CONDUCTING CYLINDERS, such as were partly secured by Smeaton by the use of his wood-lined pistons and heads, and such as have since been sought by Emery and others; such as was shown to be needed by Watt, and later more conclusively by Rankine and his successors, would do away with the necessity of compounding on the ground of thermodynamic gain; but would leave the advantages of the multi-cylinder engine, on the score of better division of stresses and work, unaffected. What may be done in this direction, it is as yet impossible to judge; but it is not likely that the device of Smeaton can be made successful at modern temperatures and pressures, or in presence of superheating; the plan of Emery of using glass, enamel, or other superficial covering of the exposed surfaces, has not yet given promise of success, and nothing as yet tried seems to give promise of meeting the requirements of the case.* The value of even an approximately non-conducting covering of such nature would be considerable for the compound engine, and very great for the simple engine; especially for the smaller sizes, in which the proportion of exposed surface is comparatively large. It is not too much to expect that some inventor may yet appear to make this, the most imperative of all needed improvements of the steam-engine of whatever type. It would render the engineer independent of all special expedients for reducing wastes internally and for thus increasing efficiency.

CLEARANCES are usually greater in the multi-cylinder than in the simple engine; but it is at once seen that the waste by clearance and the rejected steam thus utilized in any one cylinder goes to fill the clearances of the next, and thus the loss by this method of waste is divided by the number of cylinders, as in the case of other losses. It remains advisable to reduce the dead-spaces as much as is practicable in the compound engine; but the importance of this matter is less than in the case of the simple engine. Thus the adoption of a multi-cylinder engine reduces wastes of every kind, except those

* The writer has recently secured an invention devised by himself, consisting in the solution of the exposed metal surfaces, leaving the carbon of the casting to form a layer resembling vulcanized rubber which is to be saturated by drying oils, solutions of gum or other non-conductor, the covering so formed being integral with the cylinder-head or other part.

coming of increased radiation from the exterior; where the total area is, as is commonly the fact, increased, and of the friction of the engine when the number of cylinders exceeds that giving a minimum. These are, however, minor wastes.

The number of subdivisions of expansion and the number of cylinders to be introduced in series is finally settled by financial considerations. The fact that the loss by internal wastes is measured by that of one cylinder indicates that, as a matter of economy of heat, simply, there is no natural limit to the number, except that the losses by external conduction and radiation may finally more than compensate the gain by further complication. This principle is easily shown, analytically, thus:

The work performed is proportional to the quantity $1 + \log r$, and the cost of that work is proportional to the quantity $1 + a\, r^{\frac{1}{mn}}$ since the expansion in one cylinder is the nth root of the total ratio of expansion for the series; m is the index determined by the rate and method of variation of the cylinder condensation with variation of the ratio of expansion, and which is not far from $m = 2$; and a is a coefficient found by Gately and Kletsch to be about 0.2 for that special case. The cost of power, measured in terms of steam expended thermodynamically and by internal wastes, is a minimum when the quotient of the two expressions, $\dfrac{1 + \log r}{1 + a\, r^{\frac{1}{mn}}}$, above is a minimum; this is a minimum when the denominator is a maximum; and this is a maximum when the second term is a minimum, or when the value of n increases, without limit.

The question which the engineer must solve is this: How many cylinders will it pay to introduce? No general solution of the problem can be given; but it is easily solved by computation for each case as it arises. The considerations involved are the following: It may be taken as the result of general experience, in good practice, that under the best customary conditions of operation a good simple engine, working at high pressure, condensing, and at the best ratio of expansion for maximum engine efficiency, may be fairly expected to give as good a result as two pounds of fuel of satisfactory quality per horse power and per hour. Under similarly favorable conditions we may, with equal likelihood, anticipate a probability that we may obtain better work with multi-cylinder engines in somewhere about the following proportion:

Engine.	Con-sumption.	Gain, Total.	Gain, Diff.
Simple, one-cylinder	2 lbs.	..	..
Compound (double expansion)	1.6"	20%	20%
Triple expansion	1.4	30	10%
Quadruple expansion	1.25	40	10
Quintuple expansion	1.1	45	5

The first three cases are based upon what is probably ample experience; the last two are obtained by inference from the rate of progression thus established, and upon the principle above enunciated, that the loss is reduced in proportion, approximately, to the number of cylinders in series. The probable cost of adding one and another cylinder to any given type is easily ascertained by the engineer; he knows the cost of fuel and oil; the value of capital is as easily ascertained; and he can then readily determine whether the gain fairly to be anticipated is sufficient to compensate the cost of its acquirement and to give a fair margin of profit.

Another important inference from what has preceded is that the question of use of one or another type of multi-cylinder engine is not primarily settled by the magnitude of the *steam pressure* to be adopted; although it is well settled by experience and by the financial aspect of the question, as just indicated, that it will not pay to compound a machine working at very low pressures; nor to adopt a third cylinder until the pressure approaches, perhaps, four or five atmospheres, the advisability of adding cylinder after cylinder being measured by the rise in pressure, at the rate of not more than one cylinder for each four or five atmospheres pressures. Whatever the pressure, however, the compounding will divide the total thermal loss by internal wastes, approximately, by the number in series; but it does not at all follow that the efficiency of engine or the commercial efficiency will be reduced in similar ratio. On the contrary, it will never pay to carry the complication as far as the study of the ideal case would dictate. The discrepancy will be found to be the greater as the real engine the more closely approaches ideal perfection, the simple engine becoming the more desirable type as the efficiency of it and of each of the several elements of the compound engine becomes greater.

As respects size, it is now easily seen that the gain by compounding is, so far as the considerations here studied are concerned, at least, likely to prove even more marked with small than with large engines; although it may not be, commercially, as desirable to adopt this complication. As the wastes are invariably,

under similar working conditions, greater as size decreases, the desirability of reducing the magnitude of those losses would seem likely ordinarily to be made the greater, also, as size of engine diminishes. With equally dry steam from the boiler, the moisture in the steam and the losses by internal condensation are the larger as the power supplied and the magnitude of the engine furnishing it become less. That experience is showing this to be the fact is evidenced by the steady progress made by builders of small engines in the introduction of the compound engine into the market. In the case of the adaptation of this system to small engines, the effect of cylinder condensation remains in each cylinder, well marked, ordinarily, as is seen in the hitherto unnoticed effect observable where such small engines are constructed of the Wolff type; and the first effect of the cooling action of the metal upon the entering steam is shown by the sudden drop of pressure between the two cylinders, at the moment of opening communication, the fall being like that seen when exhaust occurs into the atmosphere from a high terminal expansion, and amounting, often, to several pounds.*

Problems relating to the relative efficiency of the single cylinder and the various classes of multi-cylinder engine may be readily solved, assuming the above enunciated principles to be applicable, by first computing the efficiency of the representative ideal engine, and then ascertaining the wastes, of heat, of power and of work, of the several cylinders and of each engine as a whole. Obviously, the computation of the figures for the ideal engine is precisely the same, whether, in either case, the system is simple or compound. The wastes, however, vary with each type, and with every size and proportion of engine. If, as is now possible, we may ascertain the approximate, if not exact, measure of every waste for each cylinder and for each engine, whatever its type, it is perfectly practicable to determine the relative merits of each, and the probable efficiency and consumption of heat, of steam, and of fuel also, if the efficiency of the boiler is given or calculable. The sum of the thermodynamic and the waste requirements, measures the cost of the work performed, either as the equivalent of the heat-transformation, as measured on the indicator diagram, or of the net useful work transferred through the machinery of transmission or meas-

* This has been noticed and provided for by the designers of the familiar type of single-acting compound.

ured by the Prony brake, the absorption dynamometer. The ratio of that sum to the work so measured, is the value of the efficiency of the system. The difference of efficiency among the several types or examples indicates the relative standing of those various examples and furnishes the basis for computation of the conditions of maximum efficiency of fluid, of engine, of plant, or of capital.

The following are illustrations of approximate solutions of such problems, as arising in common practice or as illustrated in the experiences of the engineer seeking to ascertain which of all available designs is the best for the special purposes in view:

First, referring to the methods of computation employed by Rankine,* we find, perhaps, the simplest and most convenient systems of treatment of the ideal case. Taking a series of values of initial pressure and of corresponding ratios of expansion, he computes the efficiencies of fluid and tabulates the results in a very compendious form. Accepting these figures, which have been checked by the writer, we have the following:

CONDENSING STEAM ENGINES WITH DRY SATURATED STEAM.

Back-pressure $p_3 \div 144$, assumed at 4 lbs. on the square inch.

Examples.	Ratio of Expansion, r, and effective Cut-off ; $\frac{1}{r}$.							
	10	5	3.33	25	2.	1.7	1.25	1.0
(1.) $p_1 \div 144 = 20.$	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0
$(p_m - p_3) \div 144$......	......	......	8.8	11.1	12.8	14 0	15.5	16.0
$p_h \div 144$............	......	.....	93	124	155	186	248	310
Efficiency of steam. ..	......	......	.095	.090	.083	.075	.0625	.052
(2.) $p_1 \div 144 = 40.$								
$(p_m - p_3) \div 144$......	......	16.2	21.9	26.2	29.6	32.0	35.0	36.0
$p_h \div 144$............	......	124	186	248	310	372	496	620
Efficiency of steam. ..	......	.131	.118	.118	.095	.086	.071	.058
(3.) $p_1 \div 144 = 60.$								
$(p_m - p_3) \div 144$......	14.8	26.3	34.9	41.4	46.4	50.0	54.6	56.0
$p_h \div 144$............	93	186	279	372	465	558	744	930
Efficiency of steam. ..	.159	.140	.125	.111	.100	.090	.073	.060
(4.) $p_1 \div 144 = 80.$								
$(p_m - p_3) \div 144$......	21.1	36.4	47.8	56.5	63.2	68.0	74.1	76.0
$p_h \div 144$............	124	248	372	496	620	744	992	1240
Efficiency of steam. ..	.170	.147	.128	.114	.102	.091	.074	.061
(5.) $p_1 \div 144 = 100.$								
$(p_m - p_3) \div 144$.....	27.4	46.5	60.8	71.6	80.0	86.0	93.6	96.0
$p_h \div 144$............	155	310	465	620	775	930	1240	1550
Efficiency of steam. ..	.177	.150	.131	.115	.103	.092	.075	.062

* "Manual of the Steam Engine," pp. 410, 411.

NON-CONDENSING STEAM ENGINES WITH DRY SATURATED STEAM

Back-pressure $p_3 \div 144$, assumed at 18 lbs. on the square inch.

Examples.	Ratio of Expansion, r, and effective Cut-off ; $\frac{1}{r}$.						
	5.	3.33	2.5	2.0	1.7	1.25	1.0
(6.) $p_1 \div 144 = 60$.	0.2	0.3	0.4	0.5	0.6	0.8	1.0
$(p_m - p_3 \div 144$.	......	......	27.4	32.4	36.0	40.6	42.0
$p_h \div 144$	......	......	372	465	558	744	930
Efficiency of steam	......	.....	.074	.070	.064	.055	.045
(7.) $p_1 \div 144 = 80$.							
$(p_m - p_3) \div 144$	.....	33.8	42.5	49.2	54.0	60.1	62.0
$p_h \div 144$	......	372	496	620	744	992	1240
Efficiency of steam	... ..	.091	.086	.080	.073	.061	.050
(8.) $p_1 \div 144 = 100$.							
$(p_m - p_3) \div 144$.	32.5	46.8	57.6	66.0	72.0	79.6	82.0
$p_h \div 144$	310	465	620	775	930	1240	1550
Efficiency of steam	.105	.100	.093	.085	.077	.064	.053
(9.) $p_1 \div 144 = 120$.							
$(p_m - p_3) \div 144$.	42.6	59.8	72.8	82.8	90.0	99.2	102.0
$p_h \div 144$	372	558	744	930	1116	1488	1860
Efficiency of steam	.115	.107	.098	.089	.081	.067	.055
10.) $p_1 \div 144 = 160$.							
$(p_m - p_3) \div 144$	62.8	85.6	103.0	116.4	126.0	138.2	142.0
$p_h \div 144$	496	748	992	1240	1488	1984	2480
Efficiency of steam	.127	.115	.104	.094	0.85	.070	.057

Taking the temperature of feed-water at such a point as will give for each case nine pounds of water evaporated into dry steam per pound of fuel, and 2.5 pounds of steam per horse-power per hour at efficiency unity, it is easy to make a comparison of the probable ideal and the probable actual efficiences of these various engines in terms of heat, steam, and fuel, demanded per unit of power in the unit of time. Rankine, in his computation as presented in the original tables, assumes an evaporation of but 7.24 per unit weight of fuel; but this is far too low to represent contemporary good practice. He also omits all correction for wastes, the two quantities to a certain extent balancing and often giving his final results in fuel consumed more nearly usual actual values than they would otherwise have exhibited. The following are selected illustrations of common practice at the several pressures and ratio of expansion given :

IDEAL EFFICIENCIES OF ENGINE.

Case No.	$p \div 144$.	r	E	Weight of (per I. H. P. per hour) lbs.	
				Steam.	Fuel.
1	20	2	0.083	30.11	3.35
2	40	2.5	0.106	23.58	2.62
3	60	3.3	0.125	20.00	2.22
4	80	4.0	0.130	19.62	2.18
5	100	5.0	0.150	16.67	1.85
6	60	2.5	0.074	33.78	3.75
7	80	3.33	0.091	27.78	3.09
8	100	5.0	0.105	23.81	2.65
9	120	5.0	0.115	21.74	2.42
10	160	5.0	0.127	18.90	2.10

Thus much for the ideal case in which the steam is either worked in a non-conducting cylinder or in an otherwise perfect engine, the steam being kept in the dry and saturated state by adding heat during expansion in just the quantity needed to prevent its partial condensation in consequence of the conversion of its heat into work. Adding to the above computed quantities of steam and of fuel those demanded to supply the wastes invariably met with in greater or less amount in all actual engines, we may obtain figures of probable approximate, perhaps closely approximate, values, for real work in the every-day practice of good engineering.

To determine the probable real efficiency of fluid, allowing for transfer without transformation, by internal wastes other than thermodynamic, assume the engines to be of moderate size and operated under familiar conditions, such as those which were met with in the experiments conducted under the system planned by the writer, by Messrs. Gately and Kletsch, in which the wastes were very exactly measured by the expression $c = 0.2\sqrt{r}$, for a non-condensing unjacketed engine, and take the losses of the jacketed engine at a common proportion, three-fourths that amount, $c = 0.15\sqrt{r}$. Adding this proportion to the previously computed amounts for the ideal case, we obtain for the actual engine figures consonant, at least more consonant, with experience. Further, assume that it is practicable, in each case, to make the mechanical efficiency of the non-condensing machine 0.90 and the condensing

engine 0.85, usual figures for the two classes. Then we obtain the following for indicated and for dynamometric power:

ACTUAL EFFICIENCIES OF ENGINE.

CASE No.	$p \div 144$.	r	E	STEAM.		FUEL.	
				I. H. P.	D. H. P.	I. H. P.	D. H. P.
1	20	2	0.069	36.2	42.6	4.0	4.7
2	40	2.5	0.085	29.2	34.4	3.2	3.8
3	60	3.3	0.098	25.5	30.0	2.8	3.3
4	80	4.0	0.100	25.0	29 2	2.8	3.2
5	100	5.0	0.109	22.9	26.9	2.5	3.0
6	60	2.5	0.050	44.9	50.0	5.0	5.5
7	80	3.3	0.067	37.0	40.1	4.1	4.5
8	100	5.0	0.073	34.2	38.0	3.8	4.2
9	120	5.0	0.080	31.3	34.8	3.5	3 9
10	160	5.0	0.087	28.7	32.0	3.2	3.6

Drier or superheated steam, higher piston-speed, larger powers of engine, efficient jacketing, will increase these efficiencies by reducing wastes; the opposite conditions will decrease them. The figures have been taken as representing fairly good practice in construction for the conditions of thermodynamic operation assumed by Rankine. Condensing engines are found to promise about twenty per cent. better performance than non-condensing, a promise fulfilled in good practice.

The differences between the steam-consumption figures of the two tables represent those wastes which may be largely reduced by compounding; they amount to a nearly constant quantity, six pounds of steam for the condensing and ten pounds for the non-condensing engines. A two-cylinder compound engine should reduce these wastes to approximately three and five pounds, a triple-expansion to two and to 3.3 pounds, a four-cylinder quadruple-expansion engine to 1.5 and to 2.5. The latter, however, with such pressures as are here assumed for the condensing engine, would unquestionably exaggerate other wastes and costs so as to, on the whole, prove unadvisable. Case No. 5, using 23 pounds of steam per hour per horse-power, would, as a compound engine, demand 20 pounds, as a triple expansion, 19 pounds, and as a quadruple expansion engine about 18.2.

All these cases, however, fail to represent modern practice; since they do not assume a sufficient expansion to give best results when compounded. The benefits of the multi-cylinder type are best seen with extreme ratios of expansion, when the internal wastes would prove excessive in the simple engine.

As a better illustration of recent and advanced practice, a quadruple expansion is to be compared with a triple expansion engine at a pressure of 200 pounds per square inch, absolute, with a back-pressure of 8 pounds and a total ratio of expansion of 16, or of 2.5^3 in the one case and of 2^4 in the other. The condenser is worked at a temperature of 150° F., in both cases, the feed being at 145° F. The friction of engine is taken in both at 15 per cent., the efficiency of machine being 0.85. The boiler evaporates nine pounds of water per pound of coal. The engines are jacketed efficiently, and the waste is taken to be measured by the factor $c = 0.15\sqrt{r}$, $= 0.15\sqrt{2.5}$ for the one case and $c = 0.15\sqrt{2}$ in the other, or 24 and 21 per cent. for the three and the four-cylinder engines, respectively. For a single engine, it would be $c = 0.15\sqrt{16}$ = sixty per cent.

Adopting Rankine's method and formulas, we obtain the following results:

For the ideal case, which would give nearly the same figures for both engines, we find the following, the slight discrepancies being due to the corresponding difference in total expansion, taking the one to work at a ratio of 2.5 for each cylinder and the other at 2:

Ideal Multi-cylinder Engine Efficiencies.

Engine.	No. Cyl.	E.	B. T. H. per I. H. P.	Water per I. H. P.	Coal per I. H. P.
Triple	1	.0811			
	2	.0730			
	3	.0779			
Total	....	.231	11761	10.85	1.35
Quadruple	1	.0637			
	2	.0598			
	3	.0580			
	4	.0598			
Total	....	.2414	11577	10.68	1.34

The consumption of water and of fuel is thus extremely low, as compared with the actual performance of the preceding cases. Adding the allowances for internal wastes, we have:

EFFICIENCIES OF REAL ENGINE.

Engine.	Water per I. H. P.	Coal per I. H. P
Ideal	10.8	1.2
Simple	17.3	1.9
Triple	13.4	1.5
Quadruple	13.1	1.4

Had these engines been unjacketed, we might probably have obtained the following by multiplying the ideal figure by $1 + c = 1 + 0.2\sqrt{r}$:

UNJACKETED ENGINES.

Engine.	Water per I. H. P.	Coal per I. H. P.
Ideal	10.8	1.2
Simple	19.4	2.2
Triple	14.3	1.6
Quadruple	13.8	1.5

The gain by increasing complication thus decreases as the number of cylinders increases, whatever the rate of internal waste.

Going into higher and unaccustomed pressures, it may be interesting to endeavor to compute the probable performance of a well designed quintuple expansion engine, working at a pressure of 500 pounds per square inch. The ratio of expansion is taken at $r = 2.3^5 = 64.4$, the back-pressure at 5 pounds. Adopting Rankine's approximate formulas, we obtain:

QUINTUPLE EXPANSION ENGINE.

Data:

$p_1 = 500 \times 144 = 72,000$ lbs. per sq. ft.;

$p_3 = 5 \times 144 = 720$;

$r = 2.3^5 = 64.4$.

Results:

$p_2 = 362.2$ lbs. per sq. ft., 6 lbs. per sq. in.

Heat expended per lb., $H = 27,324$ ft. lbs. $= 1898$ *BTU*.

$p_e = \dfrac{H}{V_2}$ || 4464 lbs. per sq. ft., 31 lbs. per sq. in.

$p_h = 17,330$ lbs. per sq. ft., 120.3 lbs. per sq. in.

Efficiency of Fluid, $E = \frac{p_e}{p_h} = 0.2576.$

BTU per IHP per hr. = 10,189.

Steam per IHP per hr., at 1100 units per lb., = 9.32 lbs.

Coal per IHP per hr. at 9 lbs. Evap. = 1.03; say 1 pound.

For this case, therefore, the weights of steam and of fuel, for unity efficiency, would be approximately 2.4 pounds, and about 0.27 pound per horse-power per hour. Were the internal wastes to be taken as in the first part of this paper, as indicated by the experiments there referred to, we should have the following, assuming the losses to be reduced in proportion to the number of cylinders employed, and the efficiency of mechanism to be 0.95 for the simple engine; 0.90, 0.90, 0.85 and 0.85 for the compounded engine in the five cases given, respectively:

EFFICIENCIES OF MULTI-CYLINDER ENGINE.

Engine.	Water per I. H. P.	Fuel per I. H. P.	E. E.	Water per D. H. P.	Fuel per D. H. P.
	lbs.	lbs.		lbs.	lbs.
Ideal Engine..........	9.32	1	1	9.32	1
Simple jacketed........	20.5	2.2	95	21.4	2 4
Double expansion......	14.9	1.6	90	16.5	1.8
Triple expansion..	13.0	1.4	90	14.4	1.6
Quadruple expansion...	12.1	1.34	85	15.0	1.7
Quintuple expansion....	11.6	1.24	85	13.6	1.5

The above is sufficient to give a fair idea, assuming our figures are satisfactorily approximate for the as yet unexplored regions to which they refer, of the advances to be anticipated by the engineer through the use of higher pressures and ratios of expansion, and with saturated steam. These figures may be decreased indefinitely by increasing boiler-efficiency and by superheating the steam.

THE INFLUENCE OF SIZE OF ENGINE may be important. In all of the examples taken, it has been assumed that the engines were of considerable size and of moderate speed of piston; at least, such that the rate of condensation found by experiment might be fairly assumed to apply to them. It will now be interesting to endeavor to obtain some idea of the effect of variation of size of engine upon their performance. That this is not necessarily serious, with even quite small engines, when proper precautions are taken to make the waste a minimum, is seen in the results of the trials of agricultural engines at the British society "Shows," where engines of

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ten and twenty horse-power are exhibited giving as high efficiency as the average of fairly good engines of the same working pressures at sea, both simple and compound being compared. But it is evident that the greater extent of surface exposed, per unit weight of working fluid subject to condensation, must, other circumstances being equal, give the larger engine the advantage.

To make this comparison it is necessary to ascertain the waste per unit area of surface exposed, per unit of time of exposure, and per unit range or temperature within the cylinder. The experiments of Messrs. Gately and Kletsch and Hill, as per Professor Marks, give for this quantity, assuming it for present purposes a constant, a value seldom far from $c = 0.02047$; which is here taken as the value affecting the cases assumed. Let the data be as follows:

Data:

Engine, single-acting compound;
Clearance, 20 per cent.;
Boiler pressure, 165 lbs. per sq. in., 23,660 per sq. ft.;
Back-pressure, 18 lbs. per in., 2,592 per sq. ft.;
Ratio of expansion in H.P. cylinder, 2.5;
Ratio of low to high pressure cylinder, 2.78 to 1;
Piston speed, 600 feet per minute;
Initial volume, v_1, 2.8 feet; final, v_2, 7 feet; $p_2 = 8,690$.

Results:

Weight of steam in low-pressure clearance, 0.554 lbs.
Compression begins at 0.047; *MEP*, in *HP* cylinder, 6,400 lbs.
Ditto in *L.P.* cylinder, 1,940 lbs. per ft.
Weight of steam in *L.P.* cylinder, 1.054 lbs.
Energy of steam per lb., 138,860 ft. lbs.
Efficiency of the steam, *E* 0.1413.
Water per *H. P.* per hour, lbs., 17.56.
Fuel at 10 lbs. per lb., 1.76.
Heat, at usual equivalent, per I.H.P. per hour, 19,766 *BTU*.

The above figures show what the ideal engine would do under the given conditions and what would be the performance of the real engine, irrespective of size, were there no wastes. With varying sizes, the volumes, v, worked at any given ratio of expansion, the stroke of piston being made variable with the diameter of cylinder, will vary as the cubes of the diameters; while the surfaces, s,

exposed will vary as the squares. The wastes occurring internally will thus vary as the quantity $s \div v$, or inversely as the diameter with cylinders of similar proportions. If the stroke be kept unchanged, the diameters varying, the wastes will vary as above, with the variation of surfaces and volumes, but less rapidly than in the first case with a given variation of power. In illustration, take three engines of the assumed type, having dimensions as below:

(1.) 18′ and 30″ × 16″ stroke;
(2.) 9″ and 15″ × 9;
(3.) 3″ and 5″ × 3″.

Taking the internal wastes, as already proposed, assuming a coefficient $c = 0.02047$, and computing the loss on the areas of the piston, the clearance, and port passages and interior of cylinder up to point of cut-off, we obtain the following results:

Variation of Efficiency with Size of Engine.

Engine.	Area s.	I.H.P.	Fuel and Water per H.P. I.H.P.	Fuel and Water per H.P. D.H.P.	Conden. perI.H.P.	Fric.
Ideal			1.76; 17.6			
No. 1	10.16	220.7	2.3 ; 23	2.7 ; 27.0	5.4	15%
" 2	2.65	30.37	2 8 ; 27.9	3.6 ; 36.1	10.30	20%
" 3	0.294	1.132	4.8 ; 48.25	6.7 ; 67.33	30.7	25%

The enormous effect of this method of waste in small engines, and the very considerable influence of size upon its magnitude in the smaller classes of engine, are thus well exhibited. It is here seen that compounding is a very much more effective means of economizing in the expenditure of fuel and of steam in small than in large engines, a remark which probably applies to all methods of increasing efficiency by reducing these once mysterious kinds of loss. In the above instances, the interior wastes increase from 5.4 pounds to 10 and to 30 pounds per I.H.P., as size decreases, and the consumption of steam thus rises from 17.6 in the ideal case, to 28 and 48 pounds for the smaller engines.

As a final illustration of the methods of treatment of the multi-cylinder engine here exhibited, an example will be selected from practice, taking an engine which is representative of the earliest attempts to employ very high steam pressures in a triple expansion engine. The steamer "Anthracite," built by the Messrs. Perkins,

is a small vessel having triple expansion engines, and boilers constructed to bear safely very high pressures. During the working and the special trials of the ship, the pressures were carried between 350 and 400 pounds by gauge; but the pressures in the cylinders fell very much below these figures. The following are data fairly representing one of the trials of this vessel, and the ideal case is worked out from them as below:

TRIPLE EXPANSION ENGINES.

Data:

$$p_1 = 201.64; \quad p_2 = 9.55; \quad p_3 = 4.21.$$
$$T_1 = 843.47; \quad T_2 = 652.2; \quad T_3 = 120.5.$$
$$v_1 = 2.236; \quad v_2 = 57.49; \quad r = 25.71.$$

Results (ideal):

$E = 0.227$; $p_2 = 3929.15$ per sq. ft. $= 27.28$ per sq. in.
Feed water per I.H.P. per hour = 8.3 pounds.

The efficiency of the boilers was, by test, 0.68, and the consumption of fuel per I.H.P. per hour, assuming it to be of such quality as would, with unity efficiency, give an evaporation of 11.75 to 1, would be, per I.H.P.:

$$8.3 \div 11.75 \div 0.68 = 0.92 \text{ lbs.}$$

The best work done on reported trials, as stated by Sir Frederick Bramwell, and as recomputed by the U. S. Naval Board repeating the trials, was 17.8 pounds of feed-water, 1.7 pounds of coal, and 20,022 B.T.U. per indicated horse-power per hour, the steam being presumed to be as above taken, dry and saturated, the jackets working efficiently. The total ratio of expansion was 25.7, in the first cylinder 2, and in the last 3.5. The waste was thus slightly more than equal to the thermodynamic demand for steam, and fifty per cent. of the total supplied. This would correspond to $c = 0.2\sqrt{r}$ for the simple engine, and to more nearly $c = 0.6\sqrt{r}$ for the single cylinder of maximum ratio of expansion. The conclusion is thus at once reached that this engine, economical as it was for its time, was not nearly as efficient as it should have been, in consequence of its wastes having been so great as to obscure the gain which should have been secured by the expansion to such a degree of such high steam. In fact, its wastes were as great as they would have probably been in a simple engine with unjacketed cylinders, working

at moderately high piston-speed. Had its superheating tubes worked with satisfactory effect, it should have been possible to reduce the expenditure of feed-water and of fuel, to as little as has been given for a similar case in the earlier part of this paper. The losses should have been not more than one-third those actually experienced, and the consumption should not have exceeded 11 or 12 pounds of water and 1.3 pounds of fuel.

How far the actual wastes were due to other methods of exaggeration of loss, as by external conduction and radiation, and by internal leakage, it is impossible to say. These may account for much of the discrepancy. Whatever the true cause, it is easy to see that a comparison of the ideal with the real case, as above illustrated, would always exhibit the fact of the waste and its amount, and enable the engineer to trace out the causes and to remedy them, either in the operation of the engine considered, or in subsequent designs, where the fault is inherent in the type or special construction.

PROBLEMS RELATING TO THE EFFICIENCY of the multi-cylinder engines may be solved most simply by the processes devised by the writer in modification of the method of Rankine, originally applied to the study of the ratio of expansion at highest efficiency of capital.* The number of cylinders or of grades of expansion being in all such cases settled by general experience and the judgment of the designing engineer, the best ratio of expansion and the best proportions of cylinders are readily determined for any given case by first obtaining the true Curve of Efficiency for the given class of engines, and then, knowing the probable back-pressure to be met with, either by custom or by taking it with reference to the best relation of initial to final pressure, and computing the constant and variable costs of operation, solving the problems, in their proper order, by a graphical construction which the writer has shown to be easily and accurately made.† It is enough to say here that these best ratios will often be found, for the better class of engines employing dry or slightly moist steam, to be not far from one-half the ratio of initial to back-pressure, the latter including the friction of engine; and for those of the very highest class, using thoroughly dry or superheated and reheated steam, on the system adopted by Cowper, Corliss, and Leavitt, this best ratio may be raised economically, on

* Miscellaneous Papers.

† The Several Efficiencies of the Steam Engine. Jour. Franklin Institute, May, 1882.

the whole, to about two-thirds the ratio of initial to back pressure. A good tentative rule is thus: *to obtain the total ratio of expansion.*

RULE.—Divide the initial-pressure in the high-pressure cylinder by twice the sum of back-pressure on the low-pressure cylinder, plus the friction of engine per unit area of piston, and the quotient will be approximately the best ratio of expansion for an average case. For the most economical classes of multi-cylinder engines, take, for the divisor, three halves the back-pressure plus friction.

It is safer, however, to endeavor to find the real curve of efficiency for the class of engine considered, and use that curve in the solution of the problems of the efficiency of fluid, of efficiency of engine, and of efficiency of plant. It thus becomes easy to ascertain the best ratios for highest duty, for best financial results as designed, as for best commercial returns should the opportunity offer of utilizing more power than is at first anticipated.

PROPORTIONS OF CYLINDERS AND RELATIVE RATIOS OF EXPANSION in the several cylinders of the multi-cylinder engine may readily be settled when the total ratio and the total power demanded are determined and exactly prescribed. It will be found that the total ratio will be made, usually, not far from equality in the several cylinders, and

$$r = r_1^n;$$

where n is the number of cylinders adopted, r the total ratio, and r_1 the ratio for one cylinder. It will, however, for best effect, on the whole, be probably advisable to adopt a compromise between the various modified and conflicting values prescribed by the conditions that the work, the effective initial pressures, and the several differences of temperature, shall be as nearly equal in all cylinders as possible. To meet the first condition we must have such a ratio in each cylinder as shall make the work in each equal to the total net power of the engine divided by the number of cylinders in series; to meet the second condition we must make the initial pressure in each such that the total range of pressure may be equal to a common range in each multiplied by the number of cylinders; while to make the range of temperature equal throughout the series, we must have varying differences of pressure, the high-pressure cylinder having the maximum range, and the low-pressure cylinder the minimum range of pressure. The differences in this

latter respect are, in engines using very high steam-pressures, quite considerable. Where the steam is dry, the speed of engine high, and the jacketing effective, this is a matter of less consequence than approximately uniform division of work and stresses on the crank-pins.

DISCUSSION.

Prof. J. E. Denton.—I notice that the subject of jackets is freely referred to in this paper, and I want to ask Professor Thurston if he will not add to his paper the experiments, as far as possible, from which he draws his conclusions. The value of it to those of us who desire to study it lies in these references. He made the statement in discussion of a previous paper that steam jackets may do no good or may do a great deal of good. Now, here is a statement that jackets are always capable of giving twenty per cent. saving. Where are the experiments that *prove* that a jacket saves twenty per cent. in modern American engines? And what were the experiments which guided him when he stated in connection with the other paper that jackets may do very little good? The philosophy referred to has been promulgated throughout the engineering world for the last forty years, and cannot be ascribed to Mr. Hirn any more than to Clark, Isherwood, Elder, Rankine, and others. But when it comes to using that philosopy to settle questions regarding ratios of expansion, proportions of cylinder, etc.—questions which I believe every expert to-day who is in actual contact with engines would hardly dare answer—I think references should be given very completely.

*Mr. Geo. H. Barrus.**—Professor Thurston's paper is exceedingly interesting to engineers and users of steam, because it deals with a great variety of mooted questions relating to the behavior of steam in steam cylinders. It is also an able paper in that it covers the whole ground of subjects pertaining to this branch of steam engineering. While being of considerable value in a popular way, it would have been of much greater value, especially to engineers, if it had dealt less in generalities, and had supplied more actual data upon the various subjects treated; and, furthermore, if some of the data given had conformed to ordinary good practice in steam engineering. I

*Added since the meeting.

dislike to bring forward adverse criticism upon the work of one of our most esteemed and influential members, and I hesitate to say even a word about it. But the object of our society makes it extremely desirable to lay aside sentiment, and to discuss the papers which are presented for just what they are worth.

A considerable portion of the paper is based on the data obtained on some experiments made by Messrs. Gately and Kletsch, two of Professor Thurston's students, which, according to a foot-note, showed that the portion of the feed water in a simple engine which is not accounted for by the indicator, and which sums up all the waste due to cylinder condensation and other losses, is a little less than 0.2 of the square root of the ratio of expansion, and the author takes this equation—that is, $c=0.2 \sqrt{r}$, as one representing the proper allowance for these sources of loss in simple unjacketed engines.

The equation given may represent the loss which occurs in some engines of a poor class, but it does not represent the loss which occurs in what may be called ordinary good practice. I do not question in the least the reliability of the experiments to which Professor Thurston refers. But I do question very gravely the fact of their representing anything but very poor practice in steam engineering. I have made a great many tests of simple engines, such as these experiments refer to, and by personal measurement have obtained the consumption of feed water per indicated horse-power per hour, and the proportion of total consumption which is attributable to cylinder condensation and leakage. In not a single case among these tests is there one to which the equation of Professor Thurston applies, unless it be that of some engine in which the test of the valves and pistons showed that so much leakage was going on as to effect largely the loss which would otherwise be attributed to cylinder condensation; and these are cases which, to my mind, should not be called upon to furnish data for any laws relating to the performance of steam engines of good ordinary type.

In the accompanying table are given the principal data and results of some of these tests upon engines, which were found to be in tight, or fairly tight, condition, as determined by trial of all the valves and pistons when the engine was at rest. These engines are all unjacketed and supplied with common saturated steam. In lines 10 and 11 appear respectively the ratio of expansion and the co-efficient *c* worked out to correspond with

Professor Thurston's equation. It will be seen that the co-efficients in these tests range from 0.10 to 0.17, the maximum, 0.17, being the case of a relatively small engine. With this individual exception, the largest co-efficient is 0.146. Leaving out this one case [which is designated B], there are five tests, viz., G, H, J, K and L, which give an average co-efficient of 0.138, and there are six remaining cases, A, C, D, E, F and I, which give an average of 0.111. The average of the whole series of twelve tests gives a co-efficient of 0.127, and this, as already noted, is a very different quantity from the one that is used in Professor Thurston's equation, viz., 0.2.

SIMPLE UNJACKETED ENGINES, WORKING WITH COMMON SATURATED STEAM; VALVES AND PISTONS TIGHT, OR FAIRLY TIGHT.

Kind of Engine.	Corliss Non-condensing.	Green Non-condensing.	Corliss Condensing.	Corliss Non-condensing.	Corliss Condensing	Corliss Condensing [same engine as C.]	Corliss ½ condensing ½ non-condensing Pair.	Corliss Non-condensing.	Corliss Condensing Pair.	Wheelock Corliss Condensing.	Corliss ½ Condensing ½ non-condensing Pair.	Corliss Condensing [same engine as K.]
Letter designating Test.	A.	B.	C.	D.	E.	F.	G.	H.	I.	J.	K.	L.
1. Diameter and stroke...	23 by 60	12 by 36	32 by 60	28 by 60	40 by 72	32 by 60	28 by 60	28 by 48	20 by 48	34 by 60	20 by 48	20 by 48
2. Revolutions per minute.............	74.7	72.7	58.6	64.8	39.6	59.1	60.3	53.2	61.7	59.9	60.3	60.3
3. Boiler pressure above atm..........	72.3	90.4	66.3	101.	50.2	70.1	61.3	69.6	84.7	82.3	69.1	67.2
4. Mean effective pressure........	33.12	42.23	39.99	41.18	25.87	38.26	½ 19.36 ½ 30.25	17.72	41.8	37.23	½ 26.41 ½ 19.01	22.79
5. Back-pressure ± atmosphere.......	+2.8	+2.3	−11.8	+4.2	−9.0	−11.6	½ +2.8 ½ −12.1	+5.9	−12.1	−13.6	½ −12.0 ½ + 0.8	−11.9
6. Feed water per indicated horse power per hour..........	27.8	29.3	19.9	25.8	24.3	19.45	25.56	34.0	19.24	18.49	22.68	21.19
7. Proportion of stroke completed at cut-off..........................	0.367	0.343	0.318	0.315	0.293	0.271	0.22	0.176	0.175	0.172	0.16	0.138
8. Proportion of feed water accounted for by indicator at cut-off........	0.84	0.71	0.81	0.82	0.79	0.78	0.72	0.70	0.74	0.67	0.70	0.65
9. Proportion of feed water not accounted for by indicator at cut-off, comprising losses by cylinder condensation and leakage............	0.16	0.29	0.19	0.18	0.21	0.22	0.28	0.30	0.26	0.33	0.30	0.35
10. Ratio of expansion.................	2.59	2.78	2.95	3.00	3.19	3.42	4.00	5.08	5.02	5.12	5.42	6.13
11. Co-efficient c in Professor Thurston's formula..............	0.10	0.174	0.11	0.104	0.117	0.119	0.14	0.133	0.115	0.146	0.129	0.141

Further evidence of the unreliability of Professor Thurston's co-efficient in showing the losses which occur in ordinary good practice may be furnished by citing the results of tests which I have made on engines which were found, on the leakage trial, to be in a more or less imperfect condition. These are all unjacketed simple engines, and, like those given in the table, are of the automatic cut-off type.

The first of these is a 23 by 60 Corliss non-condensing engine, which was working under a boiler pressure of 68.3 lbs., a speed of 74.5 revolutions per minute, and an apparent cut-off of 0.41. The leakage test showed that there was a very bad leak of the piston, which, as was afterwards discovered, was produced by a broken packing ring, and the repair of which led to a reduction of 15% in the consumption of feed water. On this test, with the engine leaking, the steam accounted for by the indicator at cut-off was 0.73, the ratio of expansion 2.33, and the co-efficient, worked out in accordance with Professor Thurston's equation, 0.179.

Another case in which leakage was found is that of a pair of 32 by 54 Corliss engines, one cylinder running condensing, and one non-condensing. The boiler pressure was 71 lbs., revolutions per minute 47.3, the average apparent cut-off 0.30, and the proportion of the feed water accounted for by the indicator 0.78. Here the ratio of expansion was 3.03, and the co efficient works out 0.123.

The case of a similar Corliss engine may be noted which was running three ends condensing and one end non-condensing, the cylinders being 26 by 60, and showing a considerable leakage of valves and pistons. The average apparent cut-off was 0.266, boiler pressure 85.1 lbs., revolutions per minute 51, and the proportion of feed water accounted for by the indicator 0 75. In this case the ratio of expansion was 3.47 and the co-efficient 0.136.

Another case is that of a Corliss non-condensing engine, 26 by 40, showing some valve leakage, which worked under a boiler pressure of 80.5 lbs., 64.7 revolutions per minute, and an apparent cut-off of 0.237. The steam accounted for by the indicator was 0.75, the ratio of expansion 3.86, and the co-efficient, worked out according to Professor Thurston's equation, becomes 0.127.

Still another case where there was a large amount of leakage going on was that of a pair of Green condensing engines, 20 by 48, which carried a boiler pressure of 60.5 lbs. The speed was

77 revolutions per minute, the apparent cut-off on one cylinder 0.29, and on the other cylinder, 0.20. The steam accounted for by the indicator here was 0.66, the average ratio of expansion 3.76, and the co-efficient 0.177.

A Woodruff & Beach non-condensing engine, with cylinder 16 by 36, boiler pressure 74.2 lbs., and revolutions per minute 75.8, working at an apparent cut-off of 0.237, furnishes an interesting example. The proportion of steam accounted for by the indicator was 0.69, and the ratio of expansion 3.76. The co-efficient works out 0.164. In this case the leakage was in the exhaust valves, and the subsequent remedy of the defect which caused it reduced the feed water consumption 10%, and brought the co-efficient down to 0.137.

Another pair of Green engines may be noted, having one cylinder 16 by 48, and one 22 by 48, which were in a bad state in the matter of leakage. This engine worked under a boiler pressure of 71.8 lbs., 94.7 revolutions per minute, and an apparent cut-off in one cylinder of 0.19, and in the other of 0.27. The proportion of feed water accounted for by the indicator was 0.67, the average ratio of expansion 3.93, and the co-efficient works out 0.168.

One other case of a leaking engine may be cited—a 16 by 42 Rollins engine, working under a boiler pressure of 75.6 lbs., a speed of 57.5 revolutions per minute, and an apparent cut-off of 0.206. Here the steam accounted for by the indicator was 0.57 of the feed water consumption, the ratio of expansion 4.36, and the co-efficient c, according to Professor Thurston's equation, becomes 0.205. This is the only instance among the eight cases of leaking engines where the co-efficient reaches the figure which has been taken by Professor Thurston to represent good practice, and this, furthermore, is the case of a relatively small engine.

I might extend these examples by referring to results of many other tests which I have made on engines of various kinds, all of which are consistent in showing the same general result. But these may suffice for the present object of my criticism as to the reliability of the data on which the author bases a large part of his paper.

The error in the co-efficient for cylinder condensation introduces a proportionate error into the figures in the table giving the "actual efficiencies of engines," which may be found on page 24 of the paper. In this table the quantity of feed water per

indicated horse-power per hour, used by an engine working under 60 lbs. pressure, condensing, is given as 25.5 lbs., and this quantity reduces to 22.9 lbs. when the boiler pressure is increased to 100 lbs. Referring to my table of results of tests with engines in good condition, it will be seen that the actual consumption of feed water per indicated horse-power per hour, where the engine is wholly condensing, varies from 18.5 lbs. to 21.19 lbs. for these ranges of pressure, and it rises to 24.3 lbs. in the case of the test marked E, made under 50 lbs. pressure. These figures are very far below the quantities given in Professor Thurston's table.

Again, taking the same ranges of pressure, viz., 60 to 100 lbs., the author gives a feed water consumption for non-condensing engines in quantities ranging from 34.9 lbs. to 42.4 lbs. per indicated horse-power per hour. Referring to my table it will be seen that the highest which is given for non-condensing engines is 34 lbs. for a pressure of 70 lbs., and the lowest 25.8 lbs. for a pressure of 101 lbs. These two are also very far below the quantities given by Professor Thurston.

While criticising these figures, I would draw attention to an evident oversight which the author has made in working out the coal consumption from the steam consumption for the two cases of condensing and non-condensing engines. The same boiler evaporation, viz., 9 lbs. of water per lb. of coal, is taken for both of these cases. This equality of the rate of evaporation is seldom realized in practice on account of the reduced temperature of the feed water in the case of the condensing engine below that which is generally obtained in the case of non-condensing engines. A comparison of the two sets of figures for condensing and non-condensing engines, given in the author's table, would therefore convey an erroneous idea as to the actual difference in economy, measured on the coal basis, between the two classes of engines.

Passing to some other questions upon which the paper treats, the author, in speaking of the reduction of the waste of fuel due to cylinder condensations, by means of superheating, advances the statement that about twenty per cent. saving is attained by effective superheating within the usual practicable range of temperature, which can be allowed in the superheater.

In my early engineering training I had much to do with this question of economy from the use of superheated steam,

being engaged on the experimental work which Mr. George B. Dixwell conducted several years ago in this field. At that time I had about the same opinion as that expressed in the author's statement, viz., that a practicable amount of superheating would secure a saving of some 20%. I would say at the outset that no such saving was ever produced in the course of Mr. Dixwell's experiments and later experiences lead me to believe that 20% is too high a figure to expect in any simple engine from the use of superheated steam.

In a series of experiments which we made on an 8 by 24 Corliss engine, running under various conditions of load and of superheating, it was found that by varying the degree of superheating according to the ratio of expansion, the saving in the quantity of steam consumed, when the superheating was carried far enough to wholly suppress cylinder condensation, amounted to about 25% for all the various conditions. This was the saving, it is to be understood, in the quantity of steam consumed and not of fuel. If account had been taken of the fuel required to superheat the steam, the saving would have been reduced a considerable amount even if there were no loss of heat in the process of superheating beyond that represented by the thermal value of the superheat.

In all the experiments of Mr. Dixwell the great difficulty in realizing much benefit from superheating seemed to be in providing a superheating apparatus which should not use up so much heat in itself as to largely offset the gain which might otherwise result.

Some positive information about the subject, conforming to what may be called modern practice, may be gained from later experiments which I have made on various large engines working with superheated steam. They were supplied from vertical boilers having a large area of steam heating surface. These engines are all comparable to those which are referred to in the table already given of engines working with common saturated steam, having been tested for leakage, and the valves and pistons found either tight or fairly tight. The results of these experiments are given in the accompanying table.

SIMPLE UNJACKETED ENGINES, WORKING WITH SUPERHEATED STEAM; VALVES AND PISTONS TIGHT, OR FAIRLY TIGHT.

Kind of Engine.	Corliss non-condensing.	Corliss non-condensing.	Corliss ½ condensing ½ non-condensing Pair.	Corliss condensing.	Corliss ½ condensing ½ non-condensing.	Corliss condensing Pair.	Corliss condensing [same engine as P].
Letter designating Test.	M.	N.	O.	P.	Q.	R.	S.
1. Diameter and stroke	23 by 60	28 by 48	23 by 60	30 by 72	34 by 60	30 by 72	30 by 72
2. Rev. per minute	74.1	100.0	61.0	54.1	53.3	46.0	54.1
3. Boiler pressure above atmosphere	79.6	68.8	80.1	53.1	83.	70.8	68.2
4. Degrees of superheating	82.	30.	59.	16.	25.	24.	16.
5. Mean effective pressure	35.07	28.6	½ 36.05 ½ 45.33	26.63	½ 28.22 ½ 41.88	32.1	26.3
6. Back-pressure ± atmosphere	+4.4	+2.8	½+ 2.6 ½+11.9	−12.4	½+ 2.1 ½−11.7	−11.5	−12.4
7. Feed water per I. H. P. per hour	26.83	26.53	21.44	19.39	22.19	18.25	18.71
8. Proportion of stroke completed at cut-off	0.392	0.331	0.281	0.247	0.233	0.185	0.165
9. Proportion of feed water accounted for by indicator at cut-off	0.95	0.87	0.89	0.82	0.77	0.82	0.75
10. Proportion of feed water not accounted for by indicator at cut-off	0.05	0.13	0.11	0.18	0.23	0.18	0.25
11. Ratio of expansion	2.44	2.85	3.31	3.72	3.91	4.79	5.29
12. Coefficient c in Prof. Thurston's formula	0.032	0.076	0.060	0.093	0.116	0.882	0.108

The average number of degrees of superheating on the seven tests given is 36, and the average proportion of feed water to represent the loss from cylinder condensation and leakage, is 0.161. The average for the saturated steam table is 0.250. Making, now, a suitable comparison between the figures representing the losses noted, it appears that a superheating of 36 degrees secured a reduction in cylinder condensation corresponding to a saving of about 10% in the quantity of feed water consumed. Before jumping at a conclusion it must be remembered that in the case of the saturated steam-engines the steam was in the condition in which steam is ordinarily supplied from boilers which do not superheat, that is, it contained a slight amount of moisture. It does not seem to me unfair to estimate that the effect of this moisture was equivalent to the effect of another 36 degrees of superheating, that is, that the saturated steam contained between one and two per cent. of moisture. In order, therefore, to make a comparison of the true effect of superheating, we must allow the equivalent of say 70 degrees of superheating for producing a saving in feed water consumption amounting to 10%. To give the superheating side of the question its full advantage and to express the relation of the two quantities in round numbers, it appears from these figures that a saving of 1.5% in the feed water consumed is produced by 10 degrees of superheating, or the equivalent of 10 degrees of superheating. Taking a temperature of 100 degrees above the normal, for the practicable limit at which superheated steam can be used, we may expect, on this basis, a saving in the feed water consumption of $\frac{100 + 30}{10} \times 1.5 = 19.5\%$; or, in round numbers, 20%.

The actual saving of fuel which can be attained out of this 20 per cent., after making sufficient allowance for the heat required to do the superheating, depends upon the efficiency of the superheating apparatus. If no heat were lost beyond that required to furnish the thermal equivalent of the superheat, the allowance would be 6.2 per cent., and would bring the net saving down to 13.8 per cent. But this, in practice, can hardly be expected, and, according to such experiments as I have made in this line, as already noted, the allowance must be a considerably greater quantity. If a vertical boiler in the ordinary form is used for the purpose of superheating, it is impossible to attain enough

advantage from superheating to balance the loss occasioned by the necessarily high temperature of the escaping gases which results. It may be possible so far to modify the ordinary vertical tubular boiler, by the introduction of water heating surface beyond the superheating surface, as to avert a large part of this waste. But how far this can be done there is no experimental proof, so far as I know, which is conclusive.

If an independent superheater is used, we must contend with the unavoidable losses due to the use of a separate furnace, with its liability to imperfect combustion and other losses, and there is the additional waste produced by the heat of the chimney gases. This system was tried in the course of Mr. Dixwell's experiments, the apparatus being well planned for the work. A careful experiment made upon it showed that about 50 per cent. more fuel was burned to produce a given amount of superheating than the quantity required, corresponding to the thermal value of the superheat. Tests of superheaters incorporated with the boiler also showed a similar loss, and it seems fair, therefore, to expect that the superheating of the steam to a temperature of 100 degrees would require the additional expenditure of about 10 per cent. more fuel over that required to generate the steam in the beginning.

From these considerations it seems to me that the effect of superheating within the range of allowable temperature, instead of being a gain of some 20 per cent., which Professor Thurston gives in his paper, does not, in reality, exceed 10 per cent., and is liable, in ordinary practice, to be less than 10 per cent.

It is greatly to be regretted that in the matter of steam jacketing, about which much is said in the paper, the author did not adduce some experimental proof of his statement that the saving due to jacketing was about 20 per cent. The statement made, that authorities disagree, is a very true one, and it would therefore have been all the more interesting if the authority for this claim had been given.

If he had also furnished some proof of the various statements as to the effect which dry or wet steam, high or low speed, simple or compound cylinders, have on the efficiency of the jacket, upon which he seems to have decided opinions, it would have done much toward answering some questions which have, for a long time, agitated the minds of steam engineers.

Mr. Frank H. Ball.—I am very much interested myself in this

subject, and I should like very much indeed to have the details of tests showing these results, as has been suggested by Professor Denton. I have made some experiments in steam jacketing in connection with compound engines, and I have not yet been able to obtain any results which compare with the figures here mentioned. I am a little unfortunate in not having had this paper earlier, or I might have perhaps brought some data with me. But inasmuch as I am not prepared with the data of my tests I will not say just what results were obtained, but they were very different from the results here described, and I hope that Professor Thurston will give us a full report of tests on which he bases these figures. I will say further that in making these tests I was very particular to measure carefully the water of condensation obtained from the jackets, and that was added to the amount of condensed steam from cylinder. I used a surface condenser for condensing the exhausts, and in making these tests I was reminded very forcibly of what we used to read about Horace Greeley's raising pork. You remember he said he made money out of pork, but he lost money out of the corn that he fed to the pork. Looking merely at the water that we obtained from the exhaust of the engine there was a great saving, but after we added the water obtained from the jacket that saving was very much reduced.

Mr. F. M. Wheeler.—There is such a difference in the matter of steam jackets that one would readily expect a marked difference in data. English engine practice (which is perhaps the largest) shows very decidedly varying results from steam jackets. There are steam jackets of many different designs and construction, and the way they are operated also makes a great deal of difference in the results.

*Prof. R. H. Thurston.**—I have read very carefully the discussion of this paper, especially the matter added since the meeting, which contains some most valuable and interesting data.

Taking the whole discussion, in the order of its presentation:

As to the first speaker: He seems to suppose that I had proposed to write a treatise on the steam jacket, and calls for references on that subject. As the writer of the paper, I may

* Author's Closure, under the Rules.

be permitted to interpret its purpose, which was simply that announced in its introductory paragraphs. It was not intended either to discuss the steam jacket at length, or to introduce any new matter in regard to it. The facts and figures given were such as I supposed all well-read engineers, making the study of the steam engine a specialty, were familiar with. I did not consider it necessary to give references which should be familiar to every such reader; and I did not imagine that others would desire more than the simple statements of one tolerably familiar with the subject. The figure 20 per cent., referred to particularly, is that which I have for many years been accustomed to take, and to see taken, as a fair one for good work with the older types of engine.

The writer of a paper may be presumed to be the best judge of the extent to which references are needed for the purposes which he has in view; and, in this case, I gave all that I thought important for such purposes: but it will give me very great satisfaction, should I find time to prepare a paper on the steam jacket, as I contemplate doing, when I shall have collected material which I am now getting together, to furnish such a complete body of data and references to authorities, both scientific and experimental, as will at least exhibit the facts of the case, whether they accord with the ideas which I have myself been forming for the last thirty years or not. I take it that no honest man will be frightened by the facts, and that no fair man will seek either to conceal or distort them.

A note has been added by a speaker, to the effect that this philosophy "has been promulgated through the engineering world for the last forty years, and cannot be ascribed to M. Hirn, any more than to Clark, Isherwood, Elder, Rankine and others."

I hardly know what is intended by this, as I have attributed to Hirn, in this paper, absolutely nothing which had not been hitherto universally admitted to be due him, except the single so-called "Hirn's Principle" defined on page 143. I have not met with that in the writings of any other recognized authority, and should be pleased to make the correction, should it be found that this, which I regard as an important principle, should be found to have been enunciated by any other and earlier writer. As to the work of the writers mentioned: I am sure no one has ever been more hearty than I in the acknowl-

edgment of our indebtedness to those great engineers. I doubt if any one has more fully exhibited their relative and chronological positions in the development of the modern theory of the steam engine.

The facts are the following: Clark experimented on locomotives as early as 1851–'52, and showed the presence in that type of engine of those causes of loss of efficiency which had been first discovered by Watt, nearly a century earlier. Clark was the first to give the "philosophy" of the phenomena so revealed. Hirn began his experimental work a little later, publishing his first paper, if I mistake not, in 1855, and giving, for the first time, ample quantitative data. Then came the work of Mr. Isherwood (1860), who, for the first time, exhibited the method of variation of cylinder wastes with variation of the ratio of expansion, thus finally developing the *law*, of which the earlier work of his predecessors had located points in its curve.

Mr. Ball's experimental work accords perfectly with the experience of earlier and well-known investigators, and simply confirms what I have said in the body of my paper, and what I will endeavor fully to justify in a later paper, should I find time to prepare it. Mr. Wheeler, who follows him in the discussion, is, I think, perfectly correct in his statements, and they fully account for the discrepancy between the results obtained by Mr. Ball and the figure which he seems to have supposed—erroneously, of course—that I had attributed to engines working under such conditions as did his own. As I have stated in the body of my paper, "The effect of a steam jacket depends upon the conditions of operation of the engine largely, and it may be productive of marked advantage, or, under unfavorable conditions" —as to its economical working, of course—"of no important useful effect."

The discussion of the paper added since the meeting by Mr. Barrus is full of matter of actual value. Before taking it up I must repeat my statement, that I had not intended to write a treatise on the steam jacket, had only touched upon that matter incidentally, and had only given what I considered to be well established and generally accepted facts and figures—and not many of them. The same is true of the main point considered by Mr. Barrus. I assumed the particular figures to which he takes exception, not as important at all for my purposes as

numerical values, but simply to illustrate a *method* which, so far as I am aware and so far as the evidence goes, up to date, is new, and which method I thought might possibly prove interesting, and, ultimately, when all the required facts and data are settled satisfactorily, correspondingly useful. The figures obtained from the Gately and Kletsch experiments were, as I well knew, accurate *for that case*, and, whether later improved upon or not, answered every purpose at the moment in view.

Mr. Barrus objects to these numerical data as not representing modern good practice, and presents figures which he considers more in accordance with such practice. His apology for their presentation is entirely unnecessary, so far as I am concerned. A chronic fault finder is a nuisance, and should be promptly abated; but a fair discussion, courteous in tone and abounding in facts, as does his paper, is always welcome to any one seeking only new and useful truths.

He would have definite figures in place of "generalities." The paper is merely an introduction of the subject. Plenty of data are available for use at a proper time, and will be given, probably. For the purposes of my paper, their source being understood and the conditions being known under which they were obtained, either the constants taken or the newer figures given by Mr. Barrus would be equally valuable and useful. I should be inclined to disagree with Mr. Barrus, however, in his classification of the two sets. I should say that the work of the first investigators illustrates fairly good practice, as average modern engineering goes—as it was desired that it should—while I should think the work of Mr. Barrus' representative of unusually good practice. However that may be, the fault is on the right side, in my view. Either set of figures, those given by me or those by Mr. Barrus, would equally well answer my purpose in this case. Mr. Barrus will find the "*constant*" reduced in numerical value by every condition conducing to the economical operation of engines. Taking his values, it means simply that the wastes, in such sizes and classes of engines as he is accustomed to working with, are apparently less than in the cases which I have assumed as representative. It is easy to apply the method which I have illustrated to such sizes and classes of engines, and it may be fairly presumed that the deductions will be found, for such practice, equally consonant with the actual case. It must be remembered, however—and this is probably

the source of Mr. Barrus' misapprehension—that the co-efficient to which he refers is only exact for cases similar to those from which its numerical values are derived. It decreases with increase in size of similar engines also, being probably nearly proportional to the reciprocal of the diameter of cylinder, and increases with the storing power of the cylinder-walls.

As the value of the Gately and Kletsch investigation, apparently, is likely to prove a matter of some importance, it may be stated here that the work was very carefully planned by me long before an opportunity offered to carry it out. It was undertaken by the two gentlemen who have reported it, one of them being able to secure the engine used, a Corliss engine of 18″ by 42″ cylinder, and to fit it up. It was a comparatively new engine, built by Harris only four years earlier, and was found to be in good order. As stated in the report, leakage "was particularly looked for, as, in these trials particularly, its existence would be fatal." None was detected. The work was done with great care, and by two men who were considered by me to be thoroughly competent as observers. One of them, in a considerable period of service, proved himself one of the most competent assistants that I ever had. They were constantly aided by advice and direction, whenever needing it, and were also assisted, at times, both by myself and by one of my assistants, Professor C. A. Carr of the U. S. Naval Engineer Corps. The reduction of observations was made under my own eye mainly, and according to a scheme which I had previously arranged. I laid out the plan of the report, and it was most skilfully and conscientiously carried out. I believe the results to be absolutely reliable. They most certainly accomplished their main purpose, the determination of the method—the law—of variation of the proportion of internal wastes with variation of the several conditions affecting them. This had never before been done, and the investigation stands to-day on record as the first of its kind, and as of extraordinary importance in the development of what M. Hirn calls the "experimental theory" of the heat-engines, or what I have called the "Theory of the Real Engine." These experiments gave a very constant value—for that engine, of course—for the quantity which Mr. Barrus has re-determined, under probably better conditions, as to efficiency, and was found to fall between 0.1740 and 0.1987 throughout the whole range of usual operation with varying ratios of expansion.

It must be remembered, however, that as just stated, this co-efficient is not a constant except for the one engine from which it is derived, and for others like it in dimensions and working condition. Other things being equal, it will be less with increasing size of engine, with increasing speed, and with increased efficiency of all those expedients which may be adopted for lessening wastes. The smaller figures obtained by Mr. Barrus are probably partly due to differences in the physical condition (as to heat-storage and transfer mainly), and partly due to differences of size and condition of the surfaces of the engine. His engines are larger than that above referred to, and we should therefore expect smaller values of that co-efficient.

Where *different sizes* of engine are to be compared, I know of no other or better method than that illustrated in the paper presented by me, in the use of what has been called "Marks' co-efficient." This will be found fairly constant, under similar conditions of internal surface, I imagine, for all types of engine, and for all sizes. Its value is found by Marks to average 0.02047 for a series of engine-trials by Hill, and to have a slightly lower value for the engine of Gately and Kletsch. All of these quantities are, however, affected by quality of steam, as well as by condition of engine. I doubt if any can be taken as more than approximate as yet, and then only under specified conditions.

I must correct the statement to the effect that it is the values of this datum "on which the author bases a large part of his paper." As already stated, the magnitude of these co-efficients is absolutely of no consequence in the construction of that paper, and the method which it is brought forward to illustrate is just as well exhibited by the use of data obtained from more, as from less, wasteful classes of engines. Again, he states that I give figures for water-consumption which he thinks high. The fact is that I simply show what would be the probable water-consumption in engines wasting heat in the proportion assumed. It is easy to similarly adjust the computations to cases in which, like those of Mr. Barrus', the wastes are lower in amount. It being understood that this quantity, which has been studied by him to such good purposes, is variable with size of engine and other conditions, all to be defined and identified for any given case, his misapprehension, and the discrepancy which he has supposed to exist, disappear.

I had supposed these matters more generally understood than

they now seem to be. It is easy to substantiate, I think, every point which I have made in the paper, and a study of the literature of the subject, in English, French and German, for the last ten or fifteen years, will probably yield all the information that may be desired. Clark, Isherwood, Cotterill, Rankine, and later writers in English; Hirn, Combes, Hallauer, Dwelshauvers-Dery, and others in French; and Grashof, Schröter, and a few others in German, furnish both the experimental and the philosophical material for the "Theory of the Real Engine."

Mr. Barrus goes on to say that the result of Mr. Dixwell's experiments on superheating gave a lower value of the saving practicable by superheating than I have assumed, although, as he states, he had himself previously had an idea that my figure was about right. He says that Mr. Dixwell never obtained as much as 20 per cent. gain by superheating. Admitting that to be the fact, I must be permitted to adhere to my own conviction, as derived from a study of not only the work of Mr. Dixwell, but of many other investigators. It will afford me great pleasure, at the proper time and in the proper place, to give the facts and references that support my statement. It should be remarked that the saving of something under 25 per cent., which he at once proceeds to give as the result of experiments on a Corliss engine by Mr. Dixwell, may be taken as ample proof that, on an engine of any important size and power, the gain should be considerably more. A toy engine like that, with its enormous wastes, which, presumably, superheating could with difficulty be overcome, would be, I think, at a great disadvantage in consequence of its small size, as compared with ordinary engines such as, for example, Mr. Isherwood tested. Mr. Barrus himself at once gives us data in his table showing a gain of 10 per cent. on larger engines by a superheat of but 36 degrees—say fifteen or twenty degrees, only, above the point at which, according to Zeuner and others, the steam becomes actually and completely dry. Had the superheating been made 100 degrees, I presume that it would have been found that about as great an advantage would have been secured as I have assumed as a fair average figure. In fact, Mr. Barrus computes precisely this figure in the very paragraph which encloses his table, but computes wastes reducing it to 14 per cent. That the conditions mentioned by him as liable to modify this saving do arise, no one can doubt; but the facts are, I think it is well proven by experiment, that it

is often perfectly practicable and safe to increase efficiency 20 per cent. by superheating. That this will always be found to pay in the end, when the wear and tear of apparatus is considered, I very much doubt; that is quite another matter.

But I have under my hand a copy of Mr. Dixwell's paper of 1875, a copy of his circular letter of May, 1877, and the report on his experiments by the naval board of March, 1877. In the first, which is a most tantalizingly unsatisfactory paper so far as data go, I find very little definite information; but I *do* find that he considers it perfectly safe to carry a temperature of 500° Fahr. in the steam cylinder; that, as he says, it is thus practicable to suppress cylinder-condensation, and, in the compound engine, to gain 25 per cent. by superheating. His circular letter asserts the practicability of a gain of 20 per cent. in simple engines, and the naval document gives the cost of power in steam consumed, the steam being saturated in these experiments as from 27.66 to 33.54 pounds; while with superheated steam the figures are from 19.39 to 26.48 pounds, or a gain on the average of slightly over 20 per cent. So much for Mr. Dixwell.

In 1854 Mr. Isherwood reported to the Journal of the Franklin Institute experiments on the "Joseph Johnson," giving a gain of 50 per cent. and upward; the P. & O. Co. of Great Britain found a saving a long time ago of 25 per cent.; the British naval steamers "Black Eagle" and "Dee" gave a saving of over 30 per cent., as reported; John Bourne reports the saving by the use of the Wethered system at sea to be 23 to 34 per cent., and the work of Mr. Isherwood on the "Eutaw" and the "Georgiana," giving similar gains, is too well known to every engineer to make it necessary to detail its results here. I am not at all sure that these and the still larger figures sometimes reported are for usual or fairly representative conditions, but they are on record.

It would be equally easy, were it important to do so here, to quote evidence as to the value of the steam jacket on various classes of engine in support of my statements; but it is enough to say that we find in the Encyclopædia Brittanica an estimate from experiments of 20 to 25 per cent. for simple engines—the case to which I referred in my paper when giving a similar figure; Cornut gives 19 per cent. for a Corliss engine; Hirn gives about 20 per cent. for a pair of engines exactly alike, as he states, except in the use or the disuse of the jacket; the same figure, substantially, is given by Hallauer; Isherwood gives similar gain

for a small simple engine, and the experiments of Emery on his revenue boats, when operated as simple engines, give also a very similar figure. It would be easy to find evidence in ample amount proving a general advantage in its use on compound engines, while there are now many cases on record, going certainly as far back as Mr. Isherwood's work on the Brooklyn pumping engine, of *no* important advantage being found to accrue from its use on some engines, especially those of high efficiency due to compounding effectively, to high speed of piston, or other effective method of reducing those internal wastes which the jacket is a wasteful apparatus for checking.

I have no doubt that it would be easy to substantiate as completely every essential statement in my paper. I will see that, at least so far as the jacket is concerned, this is done at the earliest opportunity, and I anticipate that it will be found practicable to secure so much evidence that we may conclude safely that some of the more important questions of an earlier day are now beyond dispute, and that the "expert" should be able to answer them.

I am sure that all members of the society interested in this matter will agree with me that we are under great obligations to Mr. Barrus for so excellent a collection of those somewhat rare articles in this field—*reliable data.*

CCLXIII.

FLOW OF STEAM THROUGH ORIFICES.

BY C. H. PEABODY (MEMBER OF THE SOCIETY), WITH L. H. KUNHARDT (NON-MEMBER), BOSTON, MASS.

AT a former meeting of this society* an account was given of some experiments on the flow of steam in a tube 0.275 of an inch in diameter and eight inches long. The tube afforded communication between two chambers of suitable size, and the experiments consisted essentially in observing the pressures of the steam in the two chambers, and in condensing and weighing the steam after it had passed the second chamber.

If it be assumed that no heat is transmitted to or from the steam by the tube, the theory of thermo-dynamics gives for the velocity of the steam in the tube :

$$A\frac{w}{2g} = x_a r_a - x_b r_b + q_a - q_b + A\sigma(p_a - p_b), \quad . \quad . \quad (1)$$

in which A is the reciprocal of the mechanical equivalent of heat; g is the acceleration due to gravity; w is the velocity of the steam in the tube; p_a and p_b are the pressures in pounds per square foot in the reservoir and in the tube; r_a is the latent heat of vaporization, and q_a the heat of the liquid at the pressure p_a; and r_b and q_b are corresponding quantities for the pressure p_b; x_a is the part of one unit of weight of the fluid in the first chamber which is steam, $1 - x_a$ being the part that is water, while x_b is the part which is steam in the tube; finally σ is the volume of one unit of weight of water. The value of x_a is to be determined experimentally, and then x_b can be determined by the equation,

$$\frac{x_a r_a}{T_a} + \int_{t_o}^{t_a} \frac{c\,dt}{T} = \frac{x_b r_b}{T_b} + \int_{t_o}^{t_b} \frac{c\,dt}{T}, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

* Vol. X., p. 316, Transactions.

T_a and T_b being the absolute, and t_a and t_b the Fahrenheit temperatures of saturated steam at the pressures p_a and p_b. Let the cross section of the tube have the area N square feet, then the flow in pounds per second is

$$G = \frac{N w}{x_b u_b + \sigma}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

u_b being the increase of volume due to the vaporization of one pound of water.

In the application of these equations to the experiments referred to, it was assumed that the pressure in the tube was the same as the pressure in the second chamber, which gave apparently an absurd co-efficient of flow of 1.26 when the gauge pressures in the two chambers were respectively 69.1 and 4.4 pounds to the square inch. In the discussion of the paper, attention was called to the experiments and conclusions of Mr. R. D. Napier and to the review of them by Rankine.* Subsequently the writer found the very complete experiments of Fliegner† on the flow of air, in which the pressure in the tube or orifice was taken by a pressure gauge at a side orifice. At the suggestion of the writer, a similar set of experiments were made on the flow of steam, by Mr. L. H. Kunhardt of the class of 1889, Massachusetts Institute of Technology, in the preparation of his graduation thesis, from which the experimental and calculated results of this paper are taken.

The apparatus used was entirely similar to that described in the former paper, and the tests were made in the same manner, except that the pressure in the orifices was taken by aid of a side orifice, and this pressure was used as the pressure p_b in the equations given above.

The several orifices represented in section by Figs. 5, 6, and 7 were each $\frac{1}{4}$ of an inch in diameter, were well rounded to prevent contraction at the entrance, and were straight at the lower end for the distances $\frac{1}{4}$, $\frac{1}{2}$, and $1\frac{1}{2}$ inches. The side entrance in each was $\frac{1}{32}$ of an inch in diameter, and was drilled at the middle of the straight part. When in place, the orifice was screwed into a brass plate between two cast-iron reservoirs six inches in diameter and two feet long, and had a brass pipe leading

* *The Engineer*, vol. xxviii., page 359. 1869.

† *Der Civilingenieur*. vol. xx., page 14, 1874. Also, *Thermo-dynamics of the Steam Engine*, Peabody.

from the side orifice, through the second or low-pressure reservoir, to a pressure gauge at a convenient position. The brass tube was deflected to one side, so as not to interfere with the jet from the orifice, and the lower side only of the brass plate was covered with asbestos to check the flow of heat. The whole apparatus was covered with asbestos and hair felt and jacketed with Russia iron; the loss of heat by radiation was less than the probable errors of observation.

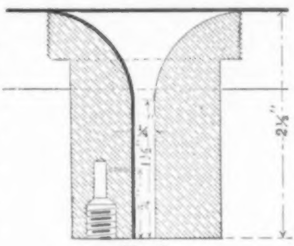

Fig. 5.

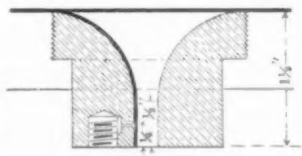

Fig. 6.

The temperature of the superheated steam in the second chamber was taken by a thermometer in a deep brass cup filled with oil, and from this temperature the condition of the steam in the first chamber could be calculated by aid of the equation

$$x_a r_a + q_a = \lambda_b + (t_s - t_b), \quad . \quad (4)$$

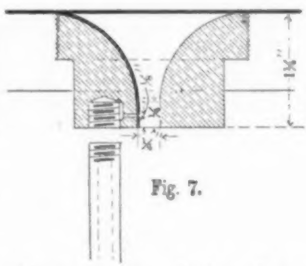

Fig. 7.

in which λ_b is the total heat of saturated steam at the temperature t_b and pressure p_b, and t_s is the observed temperature of the superheated steam in the second chamber.

Two distinct series of tests were made; those recorded in Table I. show the relation of the three pressures in the first and the second chambers and at the side orifice; those recorded in Table II. give also the amount of steam flowing through the orifice per hour.

In the first series each test is the mean of from three to five readings, taken at intervals of two minutes after the apparatus had been running at least ten minutes under nearly constant conditions. During a test the pressure never varied so much as a pound; that is, no single reading varied more than half a pound from the average reading. An examination of the four groups of experiments shows that in each group the pressure at the side orifice is affected but little by change of pressure in the second chamber till the absolute pressure in that chamber approaches half that in the first chamber. As the lower pressure approaches half the upper

pressure, the pressure at the side orifice slowly increases; but when the lower pressure becomes more than half the upper pressure, the pressure at the side orifice increases markedly with the pressure in the lower chamber.

The tests of the second series, given in Table II., were intended to be half an hour long; some were only twenty minutes long, as in the ordinary work of the laboratory it was not always easy to

TABLE I.

PRESSURE TESTS.

	Length of Tube.	Pressure above the Atmosphere, Pounds per Sq. In.			Barometer, Pounds Per Sq. In.	Ratio of Absolute Pressure in First and Second Chambers.	Ratio of Absolute Pressure at Side Orifice to Absolute Pressure in First Chamber.
		In First Chamber.	In Second Chamber.	At Side Orifice.			
	In.						
1	1 5	72.5	12.7	40.3	14.7	0.314	0.631
2	1.5	72.4	14.7	40.3	14.7	0.337	0.631
3	1.5	72.4	19.6	40.4	14.7	0.394	0.633
4	1.5	72.2	24 6	40.7	14.7	0.452	0.640
5	1.5	72 3	29.3	41.0	14.7	0.505	0.640
6	1.5	72.3	34.5	41 8	14.7	0.565	0 649
7	1.5	72.5	40.0	44.6	14.7	0.681	0.680
8	1.5	72 5	44.7	47.8	14.7	0.717	0.710
9	0.5	72.4	13.6	38.6	14.8	0.320	0.612
10	0.5	73.9	15.6	39.4	14.8	0.343	0.613
11	0.5	72.2	20.0	38 8	14.8	0.400	0.616
12	0.5	72.4	25.2	39.0	14.8	0.459	0.616
13	0.5	72.5	29.7	39.4	14.8	0 510	0.621
14	0 5	72.1	34.8	39.7	14.8	0 594	0.628
15	0.25	72.4	27.1	36.1	14.8	0.480	0.584
16	0.25	72.4	27.9	36.1	14.8	0.490	0.584
17	0.25	72.6	28.7	36.3	14.8	0.498	0.585
18	0.25	73.3	29 6	36.7	14.8	0.504	0.585
19	0.25	74.0	30.1	37.1	14.8	0.506	0.584
20	0.25	73.5	32.3	37.0	14.8	0.534	0.587
21	0.25	71.3	33.8	36.1	14.8	0.564	0.591
22	0.25	72.7	35.5	37.2	14.8	0 575	0.594
23	0.25	124.3	26.9	67.5	14.7	0 299	0.592
24	0.25	127.3	31.0	69.3	14.7	0.322	0.592
25	0.25	125.3	34.8	68.4	14.7	0.354	0.594
26	0.25	127.6	40.6	70.2	14.7	0.389	0.596
27	0 25	128.1	46.9	70.4	14.7	0.431	0.596
28	0.25	125.9	49.5	69.3	14.7	0.457	0.597
29	0.25	125.9	54.1	69.5	14.7	0.489	0.598
30	0 25	126.0	57.5	69.5	14.7	0.513	0.598
31	0 25	126.3	60.0	69.7	14.7	0.530	0.599
32	0.25	126.5	64.7	70.2	14.7	0.562	0.601

TABLE II.

FLOW TESTS.

	Length of Tube.	Duration of Test Minutes.	Pressure above the Atmosphere per Sq. In.: In First Chamber.	Pressure above the Atmosphere per Sq. In.: In Second Chamber.	Pressure above the Atmosphere per Sq. In.: At Side Orifice.	Barometer, Pounds per Square Inch.	Ratio of Absolute Pressures: Second Chamber to First Chamber.	Ratio of Absolute Pressures: At Side Orifice to First Chamber.	Temperature of Steam in Second Chamber, Deg. C.	Per cent. of Moisture in Steam in First Chamber.	Flow in Pounds per Hour: By Experiment, Weighed.	Flow in Pounds per Hour: Calculated by Thermodynamic Equation.	Flow in Pounds per Hour: Calculated by Napier's Equation.	Apparent Co-efficient of Flow.
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1..	1.5	30	74.1	14.8	41.2	14.7	0.332	0.630	126.2	0.012	221.0	217.0	224	1.018
2..	"	30	71.0	18.2	39.6	14.8	0.326	0.634	138.7	0.015	213.0	207.8	215	1.025
3..	"	20	72.6	19.7	40.6	14.7	0.394	0.634	141.4	0.005	216.0	211.4	220	1.022
4..	"	20	75.9	20.4	42.6	14.7	0.387	0.632	139.8	0.007	228.0	219.3	227	1.040
5..	"	20	71.9	24.5	40.6	14.7	0.454	0.638	140.6	0.007	213.0	209.7	218	1.016
6..	0.5	30	72.8	14.8	39.0	14.8	0.338	0.614	138.7	0.003	225.0	213.6	221	1.053
7..	"	20	72.1	20.4	38.8	14.8	0.405	0.617	142.2	0.005	223.5	211.7	219	1.056
8..	"	30	72.6	24.7	39.0	14.8	0.452	0.616	144.0	0.005	223.0	213.1	220	1.046
9..	"	30	73.1	29.9	39.2	14.8	0.509	0.615	145.2	0.005	225.5	213.0	222	1.054
10..	0.25	30	72.6	24.8	36.1	14.9	0.454	0.583	143.8	0.004	225.0	213.5	220	1.054
11..	"	30	72.6	19.9	36.1	14.9	0.398	0.583	141.6	0.004	225.0	213.5	220	1.054
12..	"	30	72.7	14.9	36.2	14.8	0.339	0.583	140.5	0.004	227.0	213.0	220	1.066
13..	0.25	30	126.3	27.8	69.0	14.7	0.295	0.594	155.0	0.005	358.8	338.9	355	1.058
14..	"	30	125.0	40.8	67.9	14.7	0.398	0.598	157.0	0.001	355.0	334.8	352	1.060

maintain a constant steam pressure for half an hour. In the thirteenth test the greatest variation from the average pressure was 2.5 pounds; in the first and third tests the greatest variation was 2.2 pounds; in all other tests the variation was less than two pounds. In the fourth, fifth, tenth, eleventh, and twelfth tests the greatest variation was not more than 0.5 of a pound. The flow of steam per hour was obtained by condensing and weighing the steam flowing from the second chamber. The calculated flow given in column 12 was found by multiplying the result obtained by aid of equation (3) by 3600, the number of seconds in an hour. In column 13 are given the results obtained by applying to these cases the equation proposed by Mr. Napier:

$$F = G\frac{p}{70}$$

with the assumption that the steam in the first chamber was dry and saturated. As determined by the temperature in the second

chamber, the priming in the first chamber was from 1 to 2 per cent. In the above equation F is the flow in pounds per second; G is the area of the orifice in square inches; p_a, the pressure in the first chamber in pounds per square inch, and 70 is an empirical divisor.

It is notable that each of the four groups of tests in Table II. shows an apparent co-efficient of flow greater than unity; that is, the actual flow is larger than that calculated by the thermodynamic equation. The co-efficient for each group is, however, fairly constant, the greatest variation from the mean for a group being less than one per cent., except in the case of the fourth test. The thermodynamic equation is deduced with the assumption that no heat is communicated to the steam during the flow. Now it is apparent that heat is communicated to the steam, since the orifice and the plate into which it is screwed are exposed to the steam at the highest pressure, and such communication of heat might be expected to increase the flow, but the extent of this action cannot now be stated. On the other hand, the pressure at the side orifice, placed at the middle of the length of the straight part of the tube, is less as the tube is shorter, while the flow and the co-efficient of flow are greater, showing an apparent friction or resistance to flow for a longer tube as compared with a shorter tube. For the shorter tubes the flow calculated by Napier's rule is in all cases less than the actual flow, but greater than the flow calculated by the thermodynamic equation. For the longest tube the flow calculated by Napier's rule is greater than the actual flow except for the fourth test, which has an exceptionally large co-efficient of flow as compared with other tests of the group.

In conclusion, it should be stated that all the tests and calculations were made by Mr. Kunhardt, who also tested, calibrated, and corrected the thermometers, gauges, and other instruments used.

[NOTE.—*This paper was presented and received discussion jointly with the author's other paper, No.* 364, *on the errors of different types of calorimeters, published on page* 193 *of Volume XI. of the Transactions.*]

CCCLXIV.

AN EXPERIMENTAL STUDY OF THE ERRORS OF DIFFERENT TYPES OF CALORIMETERS.

BY C. H. PEABODY, BOSTON, MASS.
(Member of the Society),
WITH A. L. WILLISTON
(Non-Member.)

THIS paper gives the results of some experiments on calorimeters for determining the moisture in steam, made by Mr. Williston in the laboratories of the Massachusetts Institute of Technology in the preparation of his graduation thesis. The object of the experiments was to determine the nature, magnitude, and causes of errors in calorimeters of different types, and to find how to avoid them, or how to correct them if unavoidable.

Four forms of calorimeters were used in the experiments:

(1) a throttling calorimeter designed by the writer;

(2) a Barrus continuous water calorimeter;

(3) a Hoadley calorimeter;

(4) a barrel calorimeter.

The first three types have been described in the Transactions,* and for the first, *i. e.*, the throttling calorimeter, the theoretical limits of action have been stated. It may, however, be convenient to have the following brief description.

The throttling calorimeter consists of a chamber into which steam is admitted through a throttle valve, and from which it escapes through a larger valve. The steam is superheated by the throttling or withdrawing, and its condition can be entirely determined by the temperature and pressure, which are taken by a thermometer and pressure gauge. The pressure of the steam in the main steam pipe from which the sample is taken is also observed. The calorimeter and the pipe and valve leading to it are well wrapped with non-conducting material to prevent radiation. If it be assumed that no heat is lost by the steam, the heat in one

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* Transactions, vol. X., p. 327; vol. VI., p. 297; vol. VI., p. 715.

pound will be the same in the calorimeter as in the main steam pipe, and we have the equation

$$xr + q = \lambda_c + c_p (t_s - t_c), \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which x is the part of one pound of fluid in the steam pipe that is steam, $1 - x$ being water; r is the heat of vaporization, and q the heat of the liquid at boiler pressure; λ_c is the total heat, and t_c the temperature of saturated steam at the pressure in the calorimeter; t_s is the temperature of the superheated steam in the calorimeter; and c_p is the specific heat of superheated steam at constant pressure (0.48 nearly). From the equation (1), the value of x and $1 - x$ are readily found with the aid of a table of the properties of saturated steam.

The Barrus calorimeter consists essentially of a straight vertical tube running through an open wooden bucket and forming a small surface condenser. The condensed steam is collected at the bottom of the tube, cooled and weighed. The condensing water is let in so as to circulate round the tube, is then mixed and drawn off and weighed separately. The pressure of the steam is observed, and the temperatures of the condensed water and of the cold and warm condensing water are taken. Let the weight of the condensed water be w, and of the condensing water W; let r be the heat of vaporization, and q the heat of the liquid, of steam at the pressure in the supply pipe; let q_3 be the heat of the liquid at the temperature of the water resulting from the condensation of steam, collected at the bottom of the vertical tube before it is passed through the cooler, and let q_1 and q_2 be the heats of the liquid corresponding to the initial and final temperatures of the condensing water. Then the quality of the steam x and the amount of priming $1 - x$ may be calculated by the equation

$$w(xr + q - q_3) = W(q_2 - q_1) + e, \quad . \quad . \quad . \quad (2)$$

in which e is the heat lost by external radiation, to be determined by a special experiment.

The Hoadley calorimeter is non-continuous, and consists of a surface condenser of thin copper to which steam is admitted, and in which it is collected, placed in a thin copper cylinder containing the condensing water, and thoroughly wrapped, lagged and

jacketed with eider-down and hair-felt in spaces formed by three concentric galvanized iron cylinders. Arrangements are made for stirring the condensing water and for taking the mean temperature. The calorimeter is placed on a special platform scale so that weights may be taken of the calorimeter (1), when empty; (2), when the cooling water is run in; (3), when steam has been admitted, condensed and collected in the surface condenser. The condensed water may also be drawn off and weighed separately, but not satisfactorily, as more or less water will adhere to the sides of the condenser. The equation given for this case is

$$w\,(x r + q - q_2) = (W + W')\,(q_2 - q_1) + e, \quad . \quad . \quad (3)$$

in which w and W are the weights of the condensed steam and condensing water, and W' is the water equivalent of the copper forming the condenser and calorimeter; r is the heat of vaporization, and q the heat of the liquid corresponding to the pressure of the sample of steam; q_1 and q_2 are the initial and final temperatures of the cooling water; and e is the external radiation, while x is the quality of the steam and $1 - x$ is the amount of priming.

The barrel calorimeter was a wooden barrel set on scales, with pipes and valves for supplying cold water and steam, and with a large valve for rapidly emptying the barrel at the end of a test. The temperature of the cold water was taken before it entered the barrel, and the correction for radiation and the water equivalent of the barrel avoided by first filling the barrel with hot water at about the temperature of the contents of the barrel at the end of the test, and then emptying rapidly and filling with cold water for the test. The equation for this case, using the same notation as in the preceding equations, is,

$$w\,(x r + q - q_2) = W\,(q_2 - q_1), \quad . \quad . \quad . \quad . \quad . \quad (4)$$

At the suggestion of the writer, the calorimeters were supplied with superheated steam, for which the condition could be known from the temperature and pressure, provided that Regnault's value of the specific heat c_p may be assumed to be true at all temperatures and pressures. In most of these tests the steam was superheated only a small amount to avoid uncertainty from

this source. If λ is the total heat and the temperature of saturated steam at the pressure in the supply pipe, and if t'_s is temperature of the superheated steam in that pipe, then equations (1), (2), (3) and (4) become,

$$\lambda + c_p (t'_s - t) = \lambda_c + c_p (t_s - t_c), \quad \ldots \ldots \ldots \quad (5)$$

$$w (\lambda - q_2) + c_p (t'_s - t) = W (q_2 - q_1) + e, \quad \ldots \ldots \quad (6)$$

$$w (\lambda - q_2) + c_p (t'_s - t) = (W + W') (q_2 - q_1) + e, \quad . \quad (7)$$

$$w (\lambda - q_2) + c_p (t'_s - t) = W (q_2 - q_1), \quad \ldots \ldots \quad (8)$$

(1) Throttling Calorimeter.

The errors of this calorimeter are of two sorts—those arising from inaccuracies of the thermometer or the gauge, which can be calculated, and those, like the error from radiation, which cannot be calculated.

In the description previously given, it was pointed out that small inaccuracies of thermometer or gauge had slight effect on the indications of this calorimeter. Thus at 100 pounds pressure absolute, an error of one pound in pressure giving apparently 101 pounds will make the priming appear to be from 0.02 to 0.03 of a per cent. larger than it really is. At 50 pounds absolute the error may be twice as much. On the other hand, an error of one pound in the reading of the calorimeter gauge may give an error of 0.2 of one per cent. when the pressure is 5 pounds absolute, and an error of 0.025 when the pressure is 35 pounds absolute. An error of one degree in the thermometer may give an error of 0.025 of one per cent.

Two series of tests were made on this calorimeter using superheated steam; the first, for which the results are given in Table I., to determine the effect of varying degrees of throttling and superheating, and the second, for which the results are given in Table II., to determine the effect of running varying amounts of steam through the calorimeter.

TABLE I.

THROTTLING CALORIMETER.

	Pressure absolute in supply pipe, pounds.	Pressure absolute in calorimeter, pounds.	Temperature F. in supply pipe.	Temperature F. in calorimeter.	Superheating in supply pipe.	Superheating in calorimeter.	Loss in thermal units per pound of steam used.
1	80.2	19.9	316.2	280.2	4.2	52.4	2.6
2	80.6	23.5	317.4	285.2	5.4	48.4	2.5
3	80.3	28.4	318.0	288.1	5.9	40.9	3.0
4	79.8	32.7	318.4	290.5	6.8	35.3	3.6
5	80.8	37.5	318.5	293.4	6.0	30.1	3.4
6	87.4	24.1	353.1	318.0	35.1	100.0	6.8
7	87.4	29.2	350.6	342.0	32.6	93.3	8.0
8	87.2	34.5	350.1	344.0	32.3	85.7	7.4
9	87.2	39.3	348.8	345.4	31.0	79.3	7.5

The first five tests were made with just enough superheating in the supply pipe to determine the quality of the steam, and with as nearly as possible the same quantity of steam flowing through the calorimeter per minute.

The last four tests were made with a considerable amount of superheating in the supply pipe, and with also a varying degree of throttling. The loss of heat per pound is about double that in the preceding set of tests, but this is due to the fact that less steam per minute was run through the calorimeter; the radiation, though larger, could not have been twice as great as in the preceding case. Each set of tests, considered by itself, shows that the effect of varying degrees of throttling has a very small influence on the result, the greatest variation in the loss for either group being 1.2 thermal units, or about one-tenth of one per cent.

TABLE II.

THROTTLING CALORIMETER

	Pressure absolute in supply pipe, pounds.	Pressure absolute in calorimeter, pounds.	Temperature, F., in supply pipe.	Temperature, F., in calorimeter.	Superheat in supply pipe.	Superheat in calorimeter.	Weight of steam used per hour, pounds.	Loss in thermal units per pound of steam.	Loss in thermal units per hour.	Loss per hour per degree of difference of temperature between calorimeter and air.	Error in priming from loss by radiation, per cent.
	1	2	3	4	5	6	7	8	9	10	11
1	88.0	25.0	429.8	361.0	111.3	121.0	43 5	19.3	850	2.4	2.2
2	88.0	24.9	410.5	353.1	92.0	113.3	54 0	12.8	690	2.0	1.5
3	87.0	25.0	390.2	349.2	72.5	109.2	69.0	6.4	442	1.5	0.7
4	87.4	24.9	390.2	350.2	72.2	110.4	70.5	5.5	388	1.3	0.6
5	87.5	24.9	384.8	346 1	66.7	106.3	75.0	4.9	368	1.2	0.5
6	86.1	24.9	374.4	340.5	57.5	100.7	90.0	2.8	352	0.9	0.3
7	85 7	25.0	361.9	329.2	45.3	89.2	102.0	2.3	233	0.8	0.2
8	85.2	24 9	350.4	320.7	34.2	80.9	135.0	0.9	122	0.4	0.1
9	84.0	25 0	343.6	314.4	28.4	74.4	137.0	0.9	123	0.4	0.1
10	78.0	25.0	330.4	303.1	20 4	63.1	201.0	0.9	181	0.8	0.1

An examination of Table II. shows that the greatest error of this calorimeter comes from radiation, and that for small quantities of steam per hour, the error may be 2.2 per cent. But the table also shows that the error may be reduced to one-tenth of one per cent. by the simple expedient of running a sufficient quantity of steam per hour. For the calorimeter used, which was four inches in diameter and ten inches long, at least 120 pounds of steam per hour should be used. As the calorimeter has been made with a simple valve for admitting steam, there is no way of knowing how much steam is used except by condensing and weighing it. In the future it is proposed that these calorimeters shall be provided with rounded orifices of suitable size for giving the proper quantity of steam per hour. The diameter of the orifice may be calculated with sufficient accuracy by the following formula given by Napier:

$$G = \frac{p}{70} F,$$

in which G is the flow of steam per second for the area F in square inches, and p is the absolute pressure above the orifice in pounds per square inch. For the calorimeter used in these tests, the orifice should be about 0.2 of an inch in diameter.

The greatest defect in this type of calorimeter has been shown to be that the steam will not be superheated by throttling if the priming is more than a certain per cent. depending on the pressure of the steam. This defect may perhaps be remedied by letting the steam flow through a priming box with provision for withdrawing and measuring the water that collects in it, from which box the steam can flow through an orifice to the superheating chamber. In such case the flow through the orifice must be determined by special experiment, as the tests on flow of steam presented at this meeting show that Napier's formula may be in error to the extent of two per cent.

(2) The Barrus Calorimeter.

With this type of calorimeter accurate instruments and a very considerable degree of care and skill are required to give good results. Thus, an error of one-hundredth of a degree F. in the temperature of the cooling water, whether cold or warm, will produce an error of one-thirtieth of one per cent. in the result. An error of one degree in the temperature of the condensed steam will affect the result by one-tenth of one per cent. It appears that a good thermometer will answer for the latter, but that for the former an exceptionally good thermometer must be used and that changes of the freezing point are to be guarded against. An error in the steam pressure is not much more serious than or the throttling calorimeter. The greatest difficulty is in determining the weight of the condensed steam and of the cooling water. An error of one-hundredth of a pound in the determination of the condensed steam will cause an error of one-tenth of one per cent. in the result in a test of twenty minutes duration. A slight difference of level of the water in the glass water gauge on the pipe draining the condensing pipe will make such a difference in weight, but this error is constant, and the per cent. of error can be reduced by a longer test. Common platform scales are commonly used for weighing the condensing water, and they are liable to an error of half a pound, which for a test of twenty minutes will give an error of fifteen one-hundredths of a per cent. in the result.

In Table III. are given the data and results of tests, twenty minutes in duration, on a calorimeter of this type when supplied with superheated steam. The small differences between the heat supplied and the heat accounted for show that the calorimeter is

capable of good work, and that it is not necessary to correct for radiation.

TABLE III.

BARRUS CALORIMETER.

	Boiler pressure, absolute, pounds.	Temperature, F., of superheated steam.	Temperature, F., of cold condensing water.	Temp'ture, F., of warm condensing water.	Temperature, F., of condensed steam.	Weight of steam, pounds.	Weight of condensing water.	Heat acquired by condensing water per pound of steam.	Heat yielded by one pound of steam.	Difference in B. T. U	Error in priming, per cent.
1........	87.1	344.3	53.26	85.58	306.9	331.5	11.72	912.7	915.1	+ 2.4	+ 0.25
2........	87.5	363.4	53.53	85.81	309.8	329 25	11 56	917.6	921.4	+ 3.8	0.4
3........	87.7	362.8	53.28	85.77	311.0	393.5	9.82	969.6	967.8	− 1.8	0.2
4........	89.3	414.1	54.83	86.86	310.8	328.5	11.114	945.2	944.4	− 0.8	0.1
5........	89.0	404.2	53.38	86.98	307.7	336.13	11.27	941.9	942.8	+ 0.9	0.1
6........	88.3	405.5	55.38	86.81	311.7	335.25	11.17	441.2	939.6	− 1.6	0.2
7........	88.8	402.1	55.19	87.03	311.5	337.0	11.36	940.8	937.9	− 2.9	0.3
8........	88.6	406.2	55.24	86 96	311.9	334.25	11.21	942.4	939.6	− 2.8	0.3
9........	89.1	412.9	55.26	86.79	310.7	335.3	11.12	445.4	943.9	− 1.5	0.2
10........	89.0	417.0	55.44	87.12	311.9	333.5	11.11	947.7	944.7	− 3.0	0.3

(3) Hoadley Calorimeter.

With this calorimeter, an error of one-hundredth of a degree in the temperature of the condensing water will cause an error of 0.04 of one per cent. in the result. An error of one pound in the weight of the condensing water will cause an error in the result of 0.5 of one per cent., while an error of one-tenth of a pound in the weight of the condensed steam may give an error of two per cent. The difficulty and uncertainty of drawing this water off and of weighing it separately, is the most objectionable feature of this calorimeter.

In testing this calorimeter with superheated steam, the supply was constant, and during the intervals of a series of tests, it was diverted into a condenser to dispose of it. The data and results of the tests are given in Table IV. The water equivalent of the copper forming the calorimeter and surface condenser was determined by Mr. Hoadley to be 17.2 pounds, and is included in the weights of condensing water given.

TABLE IV.

HOADLEY CALORIMETER.

	Pressure of steam, absolute, pounds.	Temperature, F., of superheated steam.	Superheat in supply pipe, degrees F.	Temperature, F., of cold condensing water.	Temp'ture, F., of warm condensing water.	Weight of condensing water.	Weight of condensed steam.	Gain in B. T. U. per pound of condensing water.	Gain in B. T. U. by condensing water per pound of steam.	Loss in B. T. U. by each pound of steam.	Difference in B. T. U.	Error in priming, per cent.
1....	72.3	337.1	32.15	54.44	78.60	225.7	4.30	22.15	1155.2	1143.8	− 11.4	1.2
2....	81.2	353.8	41.0	58.32	81.65	227.0	4.60	23.29	1149.7	1147.4	− 2.3	0.2
3....	70.1	333.0	30.2	59.74	85.56	225.2	5.10	25.78	1139.5	1135.2	− 4.3	0.5
4....	84.6	345.3	29.6	58.86	84.80	224.0	5.15	25.89	1130.0	1139.5	+ 9.5	1.0
5....	76.3	344.8	36.3	62.03	79.22	227.6	3.40	17.16	1145.3	1146.0	+ 0.7	0.1
6....	77.5	348.4	38.8	58.70	85.05	224.6	5.19	26.29	1139.9	1141.8	+ 1.9	0.2
7....	76.5	346.6	37.9	59.41	83.21	226.4	4.75	23.75	1132.0	1142.9	+ 10.9	1.2
8....	76.0	345.7	37.4	59.81	84.47	224.5	4.80	24.72	1150.1	1141.3	− 8.8	1.0
9....	76.5	346.0	37.3	59.40	90.20	224.8	6.05	30.72	1139.8	1135.5	− 4.3	0.5
10....	78.0	350.2	40.1	59.71	80.30	227.3	5.32	26.53	1135.7	1141.3	+ 5.6	0.6

(4) Barrel Calorimeter.

The tests given in Table V. were made on a barrel calorimeter for the sake of comparison with the tests on the calorimeters of other types. They were made with care and may be considered to give the degree of accuracy attainable in engineering work.

TABLE V.

BARREL CALORIMETER.

	Pressure absolute in steam pipe, pounds.	Temperature, F., superheated steam.	Superheat of steam in supply pipe, degrees F.	Temperature of cold water, degrees F.	Temperature of warm water, degrees F.	Weight of condensing water, pounds.	Weight of condensed steam, pounds.	Gain of B. T. U. per pound of condensing water.	Gain in B. T. U. per pound of steam by condensing water.	Loss of B. T. U. by each pound of steam.	Difference B. T. U.	Error in priming, per cent.
1		351.9	39.2	56.50	91.50	312.6	9.5	24.61	1143.0	1136.6	− 6.4	0.7
2		338.9	32.7	56.43	91.20	311.8	9.6	24.54	1122.8	1131.8	+ 9.0	1.0
3		347.6	36.2	56.38	88.50	320.0	9.1	24.49	1127.4	1138.9	+ 11 5	1.3
4		348.8	39.5	56.42	88.95	325.9	9.2	24.53	1146.7	1139.2	− 7.5	0.8
5		345.2	37.4	56.45	88.05	317.3	8.7	24.56	1150.3	1137.7	− 12.6	1.4
6		347.6	39.1	56.49	90.40	524.2	9.5	24.60	1154.8	1136.1	− 18.7	2.1

DISCUSSION.

Mr. A. Faber du Faur.—It appears that the tubes used in the experiments were not such as would admit of a perfect development of the jet according to the formula.

A careful investigation of the teachings of the formula would have excluded the assumption that the pressure in the tube was the same as the pressure in the second chamber; it would have shown no essential difference between the calculated values and the results of the experiments, and certainly no "apparently absurd co-efficient of flow of 1.26;" no pressure gauge at a side orifice would have been required, except to confirm the correctness of the formula.

A careful study of the formula should have indicated the form to be given to the pipes or nozzles under the various conditions of initial and final pressure.

About twenty-five years ago, while studying Zeuners' book on Thermodynamics (Freiberg, 1860), I calculated an example of flow of steam by a formula substantially the same as in the present paper. The problem was to find the velocity of flow and the diameter of pipe for one pound of steam supplied at 64.3 pounds above atmospheric pressure, and expanding without transfer of heat.

Areas relatively to smallest Areas.	Area of Orifice.	Velocity.	Diameters.	Energy Exerted.	Pressure above atmosphere per square inch.	Temperature.
∞	∞	0.	∞	0	64.3	311
1.360	1.1840	741.9	1.230	8548	54.5	302
1.090	0.9435	1051	1.090	17184	45.7	293
1.004	0.8702	1282	1.052	25645	37.8	284
1.000	0.8670	1658	1.051	42772	23.5	266
1.090	0.9648	1958	1.113	59651	14.2	248
1.310	1.1350	2222	1.202	76765	5.07	230
1.610	1.3950	2460	1.333	94019	0.00	212
2.060	1.7860	2685	1.500	110729	—4.60	194

The annexed diagram (Fig. 94) and figures show the results of

the calculations for consecutive changes of temperature of nine degrees. The abscissas are laid down by an arbitrary scale proportional to the energy exerted in producing velocity, so as to give equal work for equal distances. I have not been able to find a formula for the length of the nozzle.

An inspection of the diagram shows that after the pressure has been reduced to about 30 pounds it cannot be further reduced within the nozzle, unless the diameter of the same be enlarged. The alleged "absurd co-efficient" is fully explained by the ratios of areas of the partially developed and the fully developed jet. Between the area at 23.5 pounds and at 5.07 pounds the ratio is 1.3, or more than the alleged co-efficient of 1.26, in the calculation of which the area of the imperfectly developed jet was taken, instead of the larger area for 4.4 pounds pressure. Had the area of the pipes used in the experiments been properly contracted between the second and first chamber, no co-efficient of 1.26 would have been found.

The smallest diameter of the nozzle calculated by me is about one inch, while the smallest diameter of the nozzles used in the experiments is $\frac{1}{4}$ inch, so that at equal initial pressures the weights discharged should be as to 16 to 1. The large nozzle was calculated for a flow of 3600 pounds per hour at 64.3 initial pressure; the $\frac{1}{4}$-inch nozzle at 71 pounds initial pressure gave 215 pounds, ratio $\frac{3600}{213} = 16.9$.

Mr. H. W. Spangler.—Since the appearance of Professor Peabody's first article on his calorimeter I have been trying to repeat some of his experiments, and I must have been very unfortunate, for I have a very strong recollection that with two of those machines put on the same pipe, having superheated steam, I found nothing like the same result. I have been as careful as I knew how with my thermometers; I compared them very carefully; the gauges which I had were tested. Those of the two calorimeters were identically the same, and were especially made for this work, and I had considerable trouble in getting anything like reasonable results out of them. I was interested in the matter in the same way that Professor Peabody was, because I felt the need of something which will do this sort of work satisfactorily. I also attempted to use Barrus' calorimeter in the same way, and I sent superheated steam into those calorimeters to start with, and I found about the same

amount of difficulty with them, and the whole matter to me is in a very unsatisfactory condition.

Prof. J. E. Denton.—I think these papers are very valuable contributions to the subject of which they treat. If I understand the results aright the simple formula of Napier gives results accurate to within a couple of per cent. That is certainly something worth verifying, viz.: that one-seventieth of the absolute pressure is the flow per second into the atmosphere.

It seems to me, in regard to the calorimeters, that it is certainly a very nice idea to use superheated steam. I understand that the barrel calorimeter repeats itself within a range of two per cent. to seven-tenths of one per cent. error. As this is a subject about which there has always been a great deal of discussion, I am very glad to have a chance to agree with some one, as I can with Professor Peabody, that the errors of the barrel calorimeter are the errors of the manipulation of a complicated process. We all know that with a barrel we have made steam dry in the one test, and in the next test it has 30 per cent. of priming, and in the next it is 200° superheated. I thoroughly agree with the author that the barrel calorimeter is just as accurate an instrument as we can make it by manipulation, but that without great precaution in its use, involving much preliminary practice not usually present in our boiler tests, the errors will be excessive. But of course the barrel is not as desirable as the more modern forms of instruments.

Mr. Jas. McBride.—In the last few months I have made about forty experiments with barrel calorimeters. I weighed the water as carefully as possible on a platform scale reading quarter pounds, and used a thermometer graduated to tenths of a degree, and found I got priming from one-half of one per cent. up to slightly over nine per cent. in about forty experiments. They varied about that amount. I think I got in the whole forty experiments I made six negative results which would show superheating.*

Mr. Geo. H. Babcock.—I hope the author will permit me

* These experiments were made in two sets of twenty each, running over several weeks—some thirty days, I think. They were made under all sorts of conditions, high steam and low steam, full load and no load, high water in the boilers and low water, and heavy and light fires. The mean priming for one set was four and six-tenths per cent., and for the other three and six-tenths, so that I think, if the experiments are conducted carefully, they will give a very accurate test of the quality of the steam.

to correct an error into which he has inadvertently fallen in accrediting the formula $G = F\frac{p}{70}$ to Mr. R. D. Napier. For this very simple formula we are indebted to Prof. W. J. Macquorn Rankine, though the experiments from which it was deduced were made by Mr. Napier. Prof. Rankine called it a "rough approximation," evidently not supposing that it would be found so closely true to fact as Prof. Peabody has demonstrated. The formula proposed by Napier * was $W = 150\sqrt{\frac{P}{B}}$, in which W = weight of steam flowing from pressure, P, per minute, into a pressure $=$ or $< \frac{P}{2}$, and B the specific volume of the steam at the pressure P. This formula, though fairly approximate, is not as simple or as near correct as Rankine's. It is also true that Napier measured the pressure within the orifice in the same manner† as was afterwards done by Fleigner, and in the preparation of this paper.

Mr. Geo. H. Barrus.—Professor Peabody's reference, under the head of "Throttling Calorimeter," to the defect of this form of instrument and to its remedy by "letting the steam flow through a priming box, with provision for withdrawing and measuring the water that collects in it, from which box the steam can flow through an orifice to the superheating chamber," evidently refers to an apparatus which was devised by me a year or more ago, and to which I have given the name of "Universal Calorimeter." ‡

This instrument is shown in the accompanying cut (Fig. 98), in which A represents the drip, or priming box, and R a plate containing the orifice I. Most of the entrained water which passes through the apparatus collects in the chamber A, and the partially dry steam flows over into the chamber above the orifice, and thence into the chamber below the orifice, this last being open to the atmosphere.

The connecting unions Q and P are interchangeable, so that, if the percentage of moisture is not beyond the limitations of the wire-drawing part of the instrument, the drip box may be dispensed with. If, however, the percentage of moisture is above this limit, which, at ordinary pressures, is between three

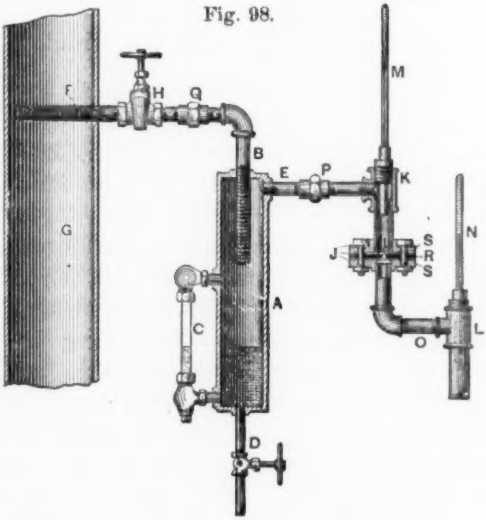
* *Engineer*, Vol. XXVIII., p. 228. † *Engineer*, Vol. XXVIII., p. 289.

‡ U. S. Pat. No. 401,111.

and four per cent., then the full apparatus, as shown in the sketch, is brought into use. It may be added here that the parts marked *J* in the cut represent insulating pieces which intercept the heat that would otherwise pass from the chamber above the orifice to the chamber below, and to some extent make the indications erroneous.

I take this opportunity to merely call brief attention to the instrument, hoping at a future meeting to give a more detailed description, and to refer to some tests upon it, made in connection with evaporative trials of boilers, which show the great

Fig. 98.



range of its action and the very interesting and valuable results which may be obtained by its use.

*Mr. C. H. Peabody.**—The kindly criticisms and corrections by Mr. Babcock are accepted gladly. It appears, however, that the formula which I have called by Mr. Napier's name, though it owes its form to Rankine, was accepted by both as representing the experiments made by the former.

In reply to Mr. Barrus, I wish to say that, while I was aware he had used a modification of the throttling calorimeter, I did not know its exact form, and that in writing the paper I had in mind particularly the errors to be guarded against if a priming box is to be used in connection with the calorimeter devised by

* Author's Closure, under the Rules.

me. Since Mr. Barrus says he intends in the future to give a detailed description of his calorimeter with an acccount of its working, I will reserve my criticisms till then.

In regard to the criticisms on the first paper, on the flow of steam, Mr. Faber Du Faur's remarks appear to me to suggest that flow of steam should always be through a tube like that represented in his figure, with an internal diameter varying according to a definite law. Experiments on such a tube would certainly be of interest. I want to call attention to the fact that the tubes used were not continually converging tubes, but were straight tubes varying in length from one quarter of an inch to one inch and a half, with well rounded openings.

In regard to the statements made by Mr. Spangler, I shall be pleased to learn, if possible, the reasons for trouble in the use of this instrument, since it appears to be a useful instrument if nothing interferes with its accuracy. Our own tests, reported in the paper, show that enough steam must be used in all cases, and that a varying and insufficient quantity may give discordant results.

As to Professor Denton's statement that our experiments verify Napier's formula, they merely show that it may be used for crude calculations; it is to be regretted that it is not more accurate.

If I understand correctly the tests made by Mr. McBride on the barrel calorimeter, the range of errors is too large to be satisfactory.

Mr. McBride.—One-half of one per cent. to about nine per cent.

Mr. Peabody.—I could scarcely agree that such results would be acceptable. I think the error should be less than two per cent. in any case, and should be less than one-half of one per cent. for results which are to be of other than merely commercial value.

CCCLXV.

ROLLING STEEL RAILS.

BY D. K. NICHOLSON, STEELTON, PA.
(Member of the Society.)

In a three high mill a rail is made at one heat, and generally in eleven passes, from a bloom seven inches square, or a little larger.

The first six passes are taken up in working the bloom from side to side into a billet, rudely, the shape of a rail. It is then ready for the finishing rolls. The seventh pass, or squabber, which is generally the first pass in the finishing train, is princi-

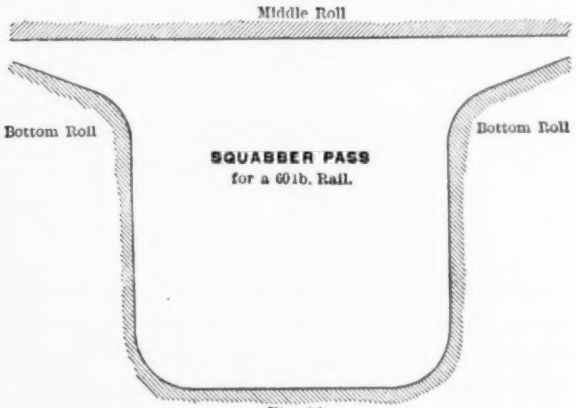


Fig. 11.

pally for forming the flanges of the rail (Fig. 11). The billet goes through this pass with the head down and the flanges horizontal; the flanges are caught between the two rolls, and made thinner and wider according to the distance between the rolls forming the pass. So that this pass has almost absolute control of the flanges. The rail passes through the three succeeding passes with head and flanges vertical, without any change except uniform reduction and

a gradual increase in the height. Then the rail goes through the finishing pass, where the head is rounded (Fig. 8), and from that it passes on to the saws to be cut.

Fig. 8.

Assuming that the passes have been properly turned out, it is the essential feature of the whole matter of rolling a rail to have them all exactly filled. If the bar does not fill out to any pass, more stuff is put in the pass; this is done by enlarging the preceding pass or passes by moving the rolls apart so as to bring out a bar of larger cross section. On the other hand, if the bar is of too great a cross section for the pass to roll out in length, the extra metal will squeeze out in the partings of the rolls, and either shear off or make a fin. The opposite course of treatment must then be resorted to.

Side-guards are used to guide the bar to the pass, and to aid in keeping it from twisting or drawing to one side on leaving the pass. Side-guards are sometimes called into play to put more metal into one side or the other of a piece by forcing it over, and compelling one side of the pass to rob the other. It is unnecessary to have a side-guard on each side of the pass when there is a greater amount of draught on one side of the bloom than on the other. The extra amount of draught on the one side throws the piece to the opposite side. There is then no need for a side-guard on the side the bar has no tendency to touch.

The purpose of a guide is to keep the bar from following the roll on coming out of the pass, when for any reason it has a tendency to do so. The pass in the rolls is turned out so as to throw the piece against the guide to insure the bar being delivered safely from the rolls. Guides and side-guards are then exactly as their names indicate, to "guide and guard" the bar in entering and on leaving a pass in the rolls.

Finning and shearing come from the metal squeezing between the partings of the rolls. It is the result, as stated above, of too much stuff going in the pass or part of the pass, or of the bar not entering the pass properly. A small fin, or the indication of one, is the only positive evidence there is that the bar has filled out as

intended. The wedge-like shape of the flange of a rail gives a considerable amount of end thrust to the rolls. If this thrusting is not met by a force sufficient to overcome it the rail will be higher on one side than the other, and have a thick and thin flange (Fig. 9).

Of all the troubles to be overcome in rolling a rail, the overfilling of the head in the finishing pass is probably the greatest. There is no real remedy for it except returning the rolls. Since the head of the rail is made somewhat rounding, the two rolls must be parted at the middle of the head in the finishing pass. When the metal runs out between the collars, it makes a fin, which is generally objected to more on account of its appearance than the harm

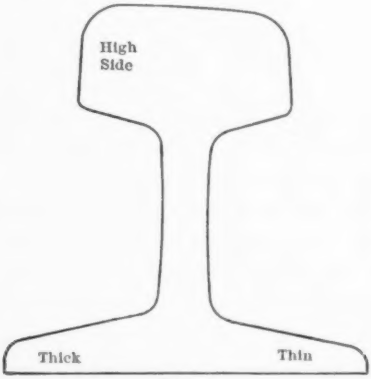


Fig. 9.

Fig. 10.

it does (Fig. 10). The cause may be looked for in the pass immediately preceding the finishing, the leading pass. When the rail has too much stuff under its head on each side of the web on coming out of this pass, on entering the finishing there is nothing to oppose this side work at the middle of the head where there is a space $\frac{1}{4}''$ wide between the collars of the rolls. From this it is plain that the greater the angle under the head of the rail the more scope for the roll turner in the leading pass, and consequently the less liability to fin or overfill in the finishing pass. Rounding the collars with a file will sometimes make the overfilling less noticeable.

In speaking on the subject of the amount of draught that ought to be put on a piece of steel, no fixed rule can be given on account of the varying conditions under which a piece is rolled. However,

taking nearly every thing into account, ten to twenty per cent. has been found to cover nearly all cases, when the piece is turned to receive work on all its sides. In breaking down a piece of steel, light draught tends to make the sides concave; the work seems to be confined near the surfaces on which it is being rolled. Heavy draught will have the opposite effect. It follows, as would naturally be supposed, that in the same rolls the hotter the piece the more the tendency of the stuff to go out in the length; while the colder and harder the piece the more spread, and consequently the more the tendency to fin. More spread may then be looked for in high carbon steel than in low carbon or mild steel. It might be here remarked that in either case the shape into which the piece is to be rolled has a good deal to do with an imperfection in the steel working out. For instance, a bad place in the part of the bloom falling to the head will work out, where it will not in the flange of a rail.

When a train of rolls is not strong enough, recourse can be had to three ways of making the rolls stronger: enlarging the diameter, shortening the body, and using better material in making the rolls. For every size of bar there is a roll of a certain diameter which will make that bar probably better than a roll of any other diameter. Of course such a thing as having a different size of rolls for every section of rails would not be practicable. So a train is selected with respect to the average work that is to be done. Rolls of small diameter are more likely to work the flaws out of a piece of steel than rolls of a large diameter. There is very little spring in a roll with a short body. For these reasons alone it appears that the second of the above-mentioned schemes (to shorten the body) would be the one to adopt.

The great drawback to a cast-steel roll is the fact that the surface cracks so badly. They do very well for roughing or where enough passes follow to smooth the bar. A forged-steel roll cracks very much less than a cast roll, but the cost puts it out of the question. As for strength, they may be said to be everlasting. This puts somewhat of a limit on the material used for a finishing roll after going outside of the best mixture of cast iron.

In a three high mill the passes in the top and middle rolls can be altered without disturbing the passes in the middle and bottom rolls. A two high mill has the advantage in handling the bar, since it enters all the passes of the rolls on the same plane. And it is only necessary to have two rolls instead of three. But then

in altering a pass, shifting one roll affects all the passes except where the finishing pass is in separate housings, which is a good thing in either train. The great speed at which the rolls are run after the bar has entered the pass in a two high reversing mill often goes against the proper formation of the rail.

In going above two lengths it is very necessary to take every precaution in putting down a mill. The long-continued strain in a set of rolls when 120 feet of rail go through is trying in the extreme on the rolls, especially as regards the end thrust. It is only by having several set screws and a well babbited surface on the lip of the brass, that the rolls can be held in their proper place in single lengths. In four lengths this would probably be double, and possibly more. Of course all this is not insurmountable if all the parts are made strong enough to resist the strain put on them, and the train kept in line,—that is, the engine shaft, the pinion and roll to which it is coupled, having their axes in one straight line, and the axes of all the rolls in the same vertical plane; for besides the train pulling hard, when the rolls are not in the same plane, the piece is liable to come out twisted.

DISCUSSION.

Mr. Robert W. Hunt.—The overfilling to which the paper refers is a point on which railroad engineers are the principal objectors: having an idea that it gives an imperfect rail. If I were having a rail made for myself, my desire would be to have a slight sign of overfilling; because it insures the fact that you had plenty of work in the last pass, and hence the maximum of work on the rail-head. Of course if the ridge is too high it makes bad wheel wear until that much steel is worn off. Personally I prefer an appearance of overfilling, but my railroad friends do not agree with that proposition. As to the material for the rolls, the author is correct in stating that a satisfactory steel roll has not been made where it has been turned for irregular shapes. A semi-steel roll has been made which in my experience has given better results than any other. It is not truly steel, but may perhaps be called a hybrid. They are very hard to turn and very expensive, costing about two and a half times as much as the best cast-iron roll. The product of rails has been brought up to such an extent that the question of price of rolls has been reduced to a minimun. For instance, if

you consider the product of the Union Steel Works of Chicago, making on one train 28,491 tons of rail in the month of October, or a 30 feet rail every 15 seconds of working hours, it only comes to about 5 cents per ton of rails. Thus the cost of rolls per ton of rails is quite a small item, and the necessity of keeping the sections true rather justifies the use of the cheaper material. Some of the mills have adopted a plan of roll-turning —I am not certain whether it originated with the Illinois Steel Company or not, but they have used it for something over a year at their South Chicago Works—of turning the rolls so that the rail enters on a slight angle. That allows them to reduce the rolls in dressing them without increasing the height of the section. This plan of roll-turning enables them to hold it to almost a uniform height until the rolls are worn out. In roll-turning, as well as in the value of the rail produced, the section plays an important part; that is, if the metal is nearly equally distributed between the head and the flange, it certainly is much easier on the train; the end thrusts are not so great, and it simplifies the roll-turner's duties, and I believe gives a much better rail. In regard to the effect of temperature upon steel, I am pretty positively committed. This heating steel to make it easy on your rolls is what has made thousands of tons of very poor rails. Hence, we must have a train strong enough to allow you to roll the steel at a moderate temperature. The statement is absolutely true that the small diameters are much easier on the steel than the larger ones. As the rail-makers increase the diameter of their rolls they increase the number of second quality rails. The question of roll-turning for rails is on its face a very simple one, but it presents some difficulties that as long as they were submitted simply to the cut and try principle of the past, gave no end of trouble to the manufacturer. But since it has been brought down to the mathematical calculations of the engineer, the mills have eliminated the greater amount of their difficulties.

Mr. Gantt.—I should like to ask about the hybrid steel roll spoken of. What kind of a roll is that?

Mr. Hunt.—I think Mr. Isaac G. Johnson of Spuyten Duyvil claims a patent on it. He makes it in an open hearth furnace, or at least in an air furnace worked with gas. He uses a large amount of rail-ends, if I mistake not, and ferro-manganese, and I think the metal contains about two per cent. carbon.

Mr. D. K. Nicholson.—There is only one remark which I want to make with regard to what Mr. Hunt says. The finishing pass is generally made so that the rail will be a little low in the beginning of the life of the roll, or, when the rolls are first started, so as to allow for the returning. The rolls may perhaps be worn out before the rail gets too high, but it sometimes happens that a set of rolls has to be thrown away because returning has so enlarged the pass as to make the height too great.

CCLXVI.

STEAM-PIPES FOR COLLIERIES.

BY E. F. C. DAVIS, POTTSVILLE, PA.

(Member of the Society.)

THE most common and the cheapest method of carrying steam, taking the world at large, is probably through wrought-iron "gas-pipe" joined by the taper thread, screwed into sockets or ferules.

This answers admirably for small pipes, and even for comparatively large pipes where the conditions are favorable for screwing up the joints, and where the threads are not subjected to any serious corrosive action. Many of the steam-pipe lines in the anthracite coal regions, however, run for great distances underground, through contracted slopes and headings where it is almost impossible to make the screwed joint. In the screwed socket joint there is always some space between the ends of the pipes, and the condensed steam from the best available feed water is so corrosive that a cutting or furrowing action takes place between the ends of the pipes and the ferule, which sooner or later causes leakage. It is then impossible to tighten up these screwed joints without screwing up the whole pipe line.

Some of these difficulties are avoided by the use of "flange unions." With these the pipe line can be more conveniently put together underground, and in the event of a leaky thread the flange can be screwed on tighter, or a defective pipe can readily be replaced by a new one of the same length. But in the ordinary flange union there is a space between the ends of the pipes, and the above-mentioned corrosive action is so destructive to the threads that cast-iron pipes have generally been considered necessary for reliable and durable steam-pipe lines; though the first cost is about double that of wrought-iron pipe.

In view of the foregoing, the Philadelphia & Reading Coal & Iron Co. has adopted the flange joint shown in annexed cut, for all colliery steam-pipes. These flanges are screwed tightly on the pipe,—the pipe carried in a steady rest,—and the end of the pipe and flange faced off flush with each other. The lugs are at the same time bored out, and the projection turned off concentric with the

bore of the pipe. This ensures perfect continuity in the pipes, and the lugs also center the gum-joint rings accurately, so that a gum-joint is obtained between the abutting ends of the wrought-iron pipes. The continuity of the bore of the pipe ensures a free flow of steam and condensed water, so that all liability to furrowing at the joints is avoided and the gum-joint formed between the ends of the wrought-iron pipes protects the thread from all danger of corrosion. If an odd length of pipe needs to be made at a colliery, the pipe—if not over four inches—can be threaded with a hand-stock and die, and a finished flange screwed on until the pipe projects through. The pipe must then be filed off flush with the face of the flange.

In moulding these flanges it is best to have the pattern arranged to leave its own cores. This ensures accuracy in the positions of the bolt-holes and the large central hole, relative to each other and to the other parts of the flange.

Several thousand feet of steam-pipe fitted with these flanges have been put in service, and have all proved perfectly satisfactory.

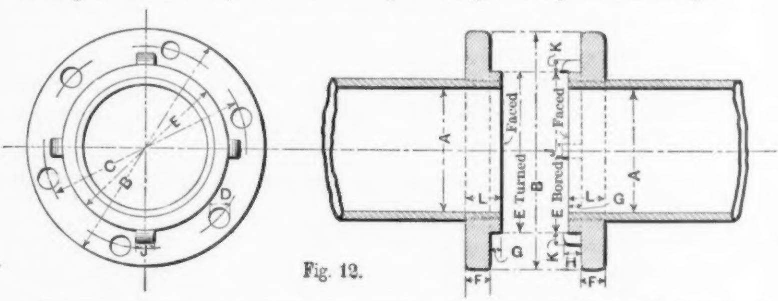


Fig. 12.

Size of Pipe.	Diameter of Flange.	No. of Holes.	⌀ of Holes.	Size of Holes.	Diagram of Facing Piece.	Thickness of Flange.	Depth of Facing Piece.	No. of Lugs.	Length of Lug.	Width of Lug.	Thickness of Lug.	Total Thickness of Flange.	Length of Pipe.
A	*B*		*C*	*D*	*E*	*F*	*G*		*H*	*J*	*K*	*L*	*Ft.*
3″	7¾″	4	6″	⅝″	5″	⅝″	¼″	4	⅞″	⅝″	¼″	1¼″	16
3½″	8½″	4	6¾″	¾″	5½″	¾″	¼″	4	⅞″	⅝″	¼″	1¼″	16
4″	9¼″	4	7½″	⅞″	6″	⅞″	¼″	4	⅞″	¾″	¼″	1⅜″	16
5″	10½″	4	8½″	⅞″	7″	⅞″	¼″	4	⅞″	¾″	¼″	1⅜″	16
6″	12″	6	10″	⅞″	8″	1″	¼″	4	⅞″	¾″	¼″	1½″	20
7″	13″	6	11″	⅞″	9″	1″	¼″	4	⅞″	¾″	¼″	1½″	20
8″	14″	6	12″	⅞″	10″	1⅛″	¼″	6	⅞″	1 ″	⅜″	1⅝″	20
10″	16¼″	8	14″	1 ″	12″	1¼″	¼″	6	⅞″	1 ″	⅜‴	1⅝″	20

Pipe must be screwed through flange so as to be steam-tight, and flange and end of pipe faced off flush at one operation.

DISCUSSION.

Mr. Carleton W. Nason.—It is a matter of surprise to me that gum packing should be used at all under steam pressure. This packing is nothing but a plastic material, a solvent making it more or less elastic, and the latter under the action of higher temperatures is speedily evaporated. The rubber then becomes non-elastic, and I cannot see how after it has been in use for a short time it should make a tight joint, and my own experience is adverse to the use of it. If the trouble, as stated in the paper, comes from the oxidation occurring at the point where the threads are exposed, I think they might be properly covered with rubber rings on the outside followed by glands, and provision could be made in case the pipes have to be afterwards cut, to use what are known as expansion joints. I should be glad to hear more fully from Mr. Davis as to the details of results.

Mr. Robert Allison.—I think there is an objection to the author's mode of putting pipes together. In the first place, in case of repairs those lugs on the flanges would prevent the removal of a section of pipe without separating the line to some extent, which is not practicable in some cases. If this is not an objection, then why not continue the lugs all the way around and make a recessed or telescopic joint which would prevent any forcing of the rings out at all under the pressure of steam. By making that recess continuous the ring would be protected all the way around, and the fitting could be done just as cheaply and would make a better job in every respect.

Prof. J. E. Sweet.—I would like to suggest to some of my friends who have the courage that they make a joint as we do on cylinder heads. It is simply to face the cylinder off in the lathe, and face the head off in the lathe, and put them together. I do not see any reason in many cases why we cannot face off the pipe ends in the lathe and bolt them together without any packing. One has no idea how easy it is to make a perfect steam-tight joint, until he tries it and does not take too much pains to make it.

The President.—The Chair hoped that Professor Sweet, when he began his remarks, was going to conclude in this fashion—that he was surprised that some members did not make experi-

ments with topics that all of us know something about, and then giving the meeting the benefit of their opinions. We would like, if possible, to have the discussion from the membership more general, especially about a matter of this kind, which is of such general interest and experience.

Mr. James McBride.—In several years' practice I have made all sorts of joints on all sorts of pipes. The principal trouble which I have found is that the flange unions on the market are not fit to make good joints. I can show numbers of flange unions broken in two, showing that the cross section of a six-inch flange union will not be much more than that of a one and a half inch. They are made entirely too light, and with too few bolts in them. I think if there was plenty of stock and enough bolts in

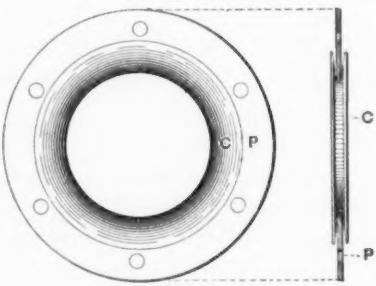


FIG. 95.

them, and these were screwed up carefully, there would not be so much trouble.

Mr. Oberlin Smith.—I believe that hearsay evidence is not always counted as valid. There is, however, a little device of which I was told the other day by one of our members which is pertinent to this subject, and which I will describe. The Standard Oil people are using it extensively for petroleum pipes; and petroleum is one of the most difficult things to make joints for. I do not know whose invention it is, or whether it is patented. Essentially it consists of a ring P, Fig. 95, of thick paper, rubber packing, or other fibrous substance, upon which is mounted a thin copper ring C, something less than one thirty-second of an inch thick, and of U-shaped section as shown. Of course this has got to be inserted while it has but one flange with the rest in the form of a cylinder, and then the part put through must be spun or otherwise formed into a flange. The beauty of it is,

they say, that the oil or whatever is in the pipe, never touches the paper or other fibre. It comes in contact with the copper only, which is elastically backed by the fibre and therefore conforms to any irregularities which there may be in the pipe flange.

The President.—The packing, then, would seem to be made by the copper, and the paper simply serves the purpose of a blocking between the copper.

Mr. Smith.—Simply to allow the copper to be elastic.

Mr. H. P. Minot.—I have noticed in the natural gas country that the men who screw the pipes together simply know enough to pull them together with a wrench. The end of the coupling should be recessed or thread taken out, and put up to a certain point, no matter how hard it goes. I believe the less packing you get in the better.

Mr. Faber du Faur.—I would mention one example of faced joints without packing on large pipes. On the Washington Aqueduct, where I was employed before the war, the pipes of Bridge No. 6, between Washington and Georgetown, are 48 inches diameter, 200 feet span and 20 feet rise, and General Meigs first intended to use packing. I insisted that the pipes should be joined without packing. They were simply faced off, painted with red lead and bolted together, and they stand well. When I first got to the Georgetown office, the Civil Engineers in charge of the office there had adopted for connecting the 48 inch pipes 7 large brackets projecting from the flanges, and 7 very thick bolts; this was changed to 42 bolts of $1\frac{1}{4}$ inch, and the joints were perfectly tight, without any packing except a thin coating of red lead.

Mr. Smith.—Does Mr. Sweet consider a perfectly tight joint, made as he says, better than the same joint with a very thin coat of linseed oil all around?

Prof. Sweet.—It is likely that the first time the oil would be all right, but the trouble is that you have to remove a cylinder head and then it has to be scraped before it is put on the next time. It must be put together absolutely dry, so that the next time you put it together it will not need any scraping, as that ruins the whole job.

Mr. O. C. Woolson.—It occurs to me if we are going to discard packing for general piping, the flanges should be male and female. They will then meet a greater want than those which

are made flush or straight across, because it is a pretty difficult matter to get the ordinary mechanic to make up a lot of piping with flange joints for a long line of pipe, which may be in position and more or less fixed, and have them tight. The packing of a joint is a sort of an expedient, and my experience is that if those flanges are made male and female, on the cup and saucer principle, they will stand a pretty good pressure and will admit of being put up by common labor. The extra expense of making such flanges is a matter of some moment however.

*Mr. E. F. C. Davis.**—Mr. Nason's objections to gum joints do not hold good in the writer's experience. A gum joint made as shown, and properly screwed up, will last in most cases as long as the pipe.

Any long line of steam-pipe should be provided with a reasonable number of expansion joints. We use the packed plunger style and with the plunger made of brass where it slides through the packing, they serve their purpose well, and give no trouble.

If Mr. Allison will bear in mind that the steam must be shut off before a pipe can be taken out, he will readily see that the shrinkage of the pipes will have a tendency to open the joint much more than the $\frac{3}{8}$ inch projection of the lugs. Practically the lugs do not interfere with repairs to a pipe line. In our first design for these flanges, the lugs were continuous, and formed a recessed or telescopic joint, as suggested. The objection to this plan is that the facing off of the end of pipe in the female flange can only be done in the lathe, while with the lugs the finished flanges can, if necessary, be screwed on at the mines, and the ends of pipes made flush with a file. This is quite a convenience in the smaller sizes, as pipe connections up to four inch diameter are frequently cut to length and threaded by hand at the mines. In sizes above four inch there is no objection to making the lugs continuous, but as no joints have blown out, the lugs answer all requirements.

By consulting the table of proportions, it will be observed that a liberal supply of large bolts is used. This is an essential feature in all joints of this kind.

Since this flange was adopted, the Philadelphia and Reading Coal and Iron Co. have put in use at their collieries:

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* Author's Closure under the Rules.

13,100	feet of	3	inch	steam-pipe,		
1,644	"	3½	"	"	"	
9,388	"	4	"	"	"	
5,190	"	5	"	"	"	
11,180	"	6	"	"	"	
2,700	"	8	"	"	"	

43,202 feet, all fitted with these flanges.

Most of this pipe is underground and all of it has proved most satisfactory.

The writer has been unable to learn of a single instance in which a joint has leaked or blown out.

CCLXVII.

INDICATOR RIGGING FOR COMPOUND ENGINES.

BY FRED. W. PARSONS, ELMIRA, N. Y.
(Member of the Society.)

It is sometimes interesting as well as instructive to combine the diagrams taken from the two cylinders of a compound engine. In order that the horizontal scale of measurements shall be the same, it is obviously necessary to reduce the high pressure diagram to a length bearing the same proportion to the low pressure diagram, as the volume of the high pressure cylinder is to that of the low pressure cylinder.

The device described is an ordinary form of indicator rigging with two segments (Fig. 13). The radius of the large segment is dependent upon the length desired for the low pressure diagram. The radius of the small segment is in the same proportion to the radius of the large segment as the volume of the high pressure cylinder is to the volume of the low pressure cylinder.

The diagram may be taken with separate indicators upon the high and low pressure cylinders, as shown in the cut, and afterward traced upon the same paper; or by piping the high and low pressure cylinders together, with a three-way cock between, both diagrams may be taken on the same paper.

It will readily be seen, that by using three and four segments, this device can be used for reducing the diagrams from triple and quadruple expansion engines.

DISCUSSION.

Mr. H. W. Spangler.—This is rather an interesting device, but I do not see the utility of it. If it was for the purpose of increasing the card instead of diminishing it, it would accomplish what most of us are after. This would reduce the card from a high pressure cylinder, where the diameter of the cylinder is one to two, from four inches down to an inch. And if it is intended that the power of the engine is to be derived from the

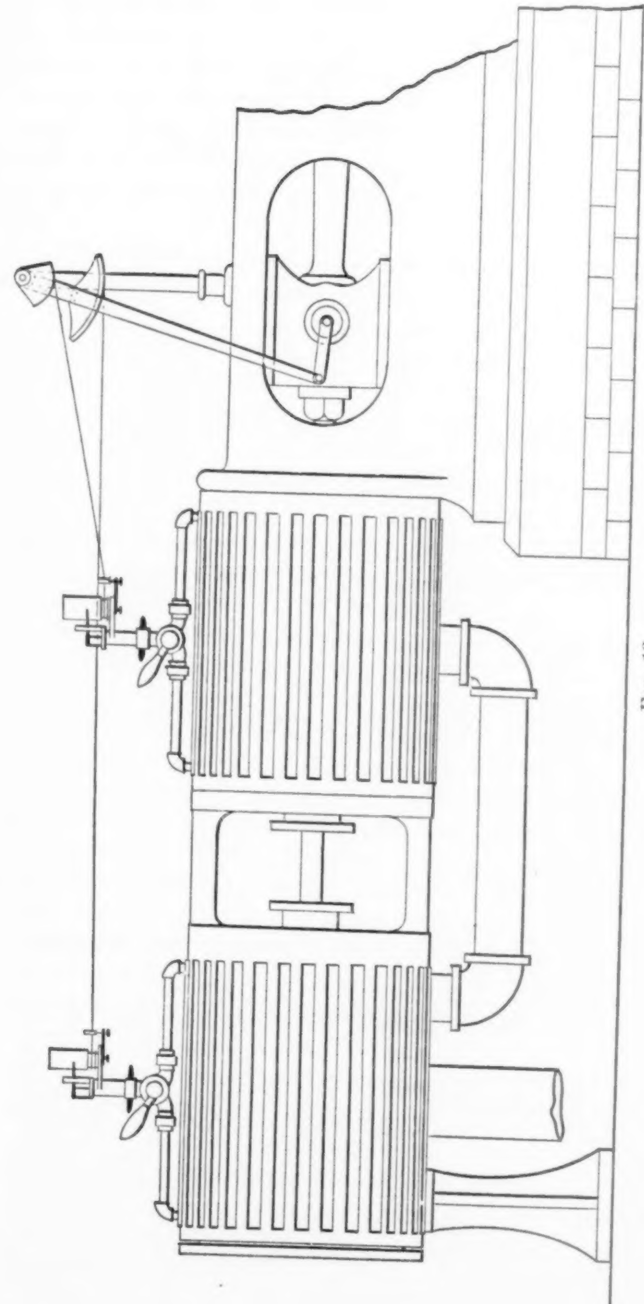

FIG. 13.

card, you have practically four times the error of the high pressure card which you had before. In very many cases it is almost absolutely necessary to enlarge the card instead of reducing it. When you come to three or four cylinders you have practically a straight line for your expansion line. For simple inspection it might give you a card which you could look at, but for practical use I do not see that it would be a device that would be of much benefit to any one.

Dr. Sellers.—This suggests an important consideration, about which I would like to say a few words. On the diagram illustrative of the device there are two different lengths of cords, reaching from the segments on the lever attached to the crosshead, or the source of motion, to the card cylinder, and the two indicators, one very much longer than the other one. A paper was read in England some time ago, and it is to be found in the Proceedings of the Institution of Civil Engineers,* on the errors which occur from the stretching of the indicator cords. In my own experience with high speed engines I found it impossible to make any connection by means of cords to get any reliable results. A rigid connection had to be used. A very high speed may be indicated with certainty if, instead of the revolving cylinder carrying the paper, a flat plate is used of wood or metal, sliding backward and forward, connected readily by rods from the lever actuated by the engine, and on this flat surface the indicator card be made.

Mr. Smith.—Those who were at the Paris Exposition may probably have seen on some of the French engines a mechanical means for connecting the cross-head directly with the indicator, so that the loss from the stretching of the string was entirely obviated. Of course there was the lost motion in that mechanism, which might be great or small, according to the mechanical construction of it; but the idea which Professor Sellers has put forward was covered in that way—at least, on one engine which I saw.

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* Osborne Reynolds on Errors in Indicator Diagrams, Vol. LXXXIII., and after Proceedings of the Institution of Civil Engineers, Great Britain.

CCLXVIII.

A NEW RECORDING PRESSURE GAUGE

BY W. H. BRISTOL, HOBOKEN, N. J.
(Member of the Society.)

In designing the recording pressure gauge herewith illustrated, the object was to produce an instrument which would be fundamentally simple and consequently reliable, and which could be placed upon the market at a moderate cost.

Fig. 14 represents the instrument complete and ready for application. Fig. 15 shows the pressure tube with the inking pointer attached; the front of case, dial, and cover of clock being removed. The pressure tube, *A*, is of flattened cross-section and bent into approximately a sinusoidal form. A flexible strip, *B*, of the same metal as the tube is secured at the ends and along the bends, as shown in Fig. 15.

The bent tube may be considered as a series of Bourdon springs placed end to end.

Pressure applied to the tube produces a tendency to straighten each bend, or collectively, to elongate the whole. This tendency to lengthen the tube is resisted by the flexible strip, *B*, and thereby converted into a multiplied lateral motion. The inking pointer is attached directly to the end of the pressure tube, as shown in Fig. 15, from which it will be seen that the usual mechanism and multiplying devices are dispensed with, since the motion of the tube itself is positive and of sufficient range. The special advantage of this is evident, considering that in all other pressure gauges the movement of the tube or diaparagm is small, and requires a system of mechanism to multiply the motion many times before it is available for indicating purposes. These multiplying devices must be delicately constructed and properly cared for, and even under the most favorable conditions they are liable at any moment to be a source of error.

In the instrument illustrated the tube is designed for a range of 180 lbs. per square inch; for other ranges its sensitiveness may be varied, at will, by changing its proportions, as length, shape

of cross-section, or thickness. The printed charts for receiving the record make one revolution in twenty-four hours, and are provided with radial arcs and concentric circles, the divisions on the radial arcs corresponding to differences in pressure; while those on the concentric circles correspond to the hours of the day and night.

During the past year and a half several of the instruments

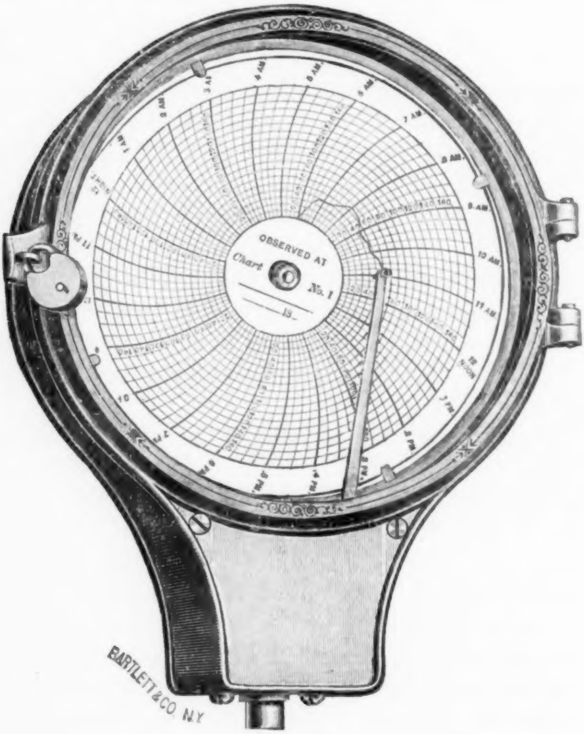

Fig. 14.

have been in operation upon the steam boilers at Stevens Institute and have given perfectly satisfactory results.

In regard to making the tubes alike, it will be well to state that there has been no difficulty in producing a number in which the deflections were equal for equal pressures, and which have been directly applied to a standard chart without adjustment. It will be readily seen that, in case there should be slight differences in the deflections, such differences may be allowed for by raising or lowering the tube with reference to the dial. This is equivalent

to shortening or lengthening the deflections along the radial arcs. For an indicating instrument it is only necessary to provide a graduated arc for the end of the tube to move over.

It is evident that the instrument is adapted for a vacuum as well as for a pressure gauge, and it naturally follows that, if sufficiently sensitive, it will serve as a barometer, and measure changes of atmospheric pressure.

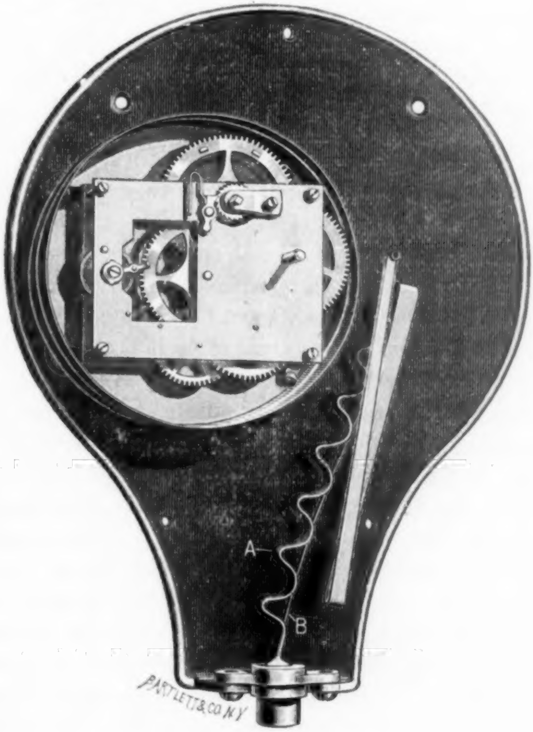

FIG. 15.

The model herewith exhibited for this purpose was made by electro-deposition of nickel upon a piece of solder of the proper form, the solder being afterward melted out in oil. The walls of this tube are $\frac{1}{500}$ in. thick. When this tube is exhausted of air and sealed, as shown, it gives a deflection of about $3\frac{1}{2}$ inches for an external change of pressure of one atmosphere.

Another application of the pressure tube is in the recording thermometer.

The tube may be filled with a very expansible liquid, such as

alcohol, and sealed. Variations in temperature produce expansion of the inclosed liquid, which, in turn, gives deflections of the tube to correspond.

These deflections may be used to record directly, without multiplying devices, as shown in one of the models.

The tubes of the pressure gauges to be inspected have been made by the writer at Stevens Institute, for the purpose of thoroughly testing the novel form. The results have been perfectly satisfactory, and our recent experience in manufacturing has demonstrated the possibility of duplicating the tubes in quantities for a standard chart.

DISCUSSION.

Mr. Chas. A. Hague.—It seems to me that this matter of recording pressure gauges is a very important one, and I have arrived at the point where I do not consider a steam plant, or a a water works plant, complete without a first-class recording pressure gauge, so that we may have on file a continuous record, which can be inspected at any time after it is made, of the pressures of steam or water that existed at any certain time.

For instance, such cases occur where boilers are left with banked fires during periods when not in use, but where it is desired to keep them in readiness to be used at short notice, or where it is desired for any reason to maintain a moderate steam pressure during the period of inactivity. When such boilers are left alone, nobody really knows to what point the pressure at times has risen, and thus seemingly inexplicable accidents happen. The pressure is carried through the day perhaps at 80 pounds, and it is supposed to be run down and kept, say, at 25 pounds all night. The fires are banked, and the boilers are left to themselves, apparently in a safe condition. But with boilers set in brickwork the fire-brick lining of the setting gets pretty hot during the day, and forms, as it were, a reservoir of heat; and if the fireman is in a special hurry to get away, it very often happens that sufficient precaution has not been taken to prevent the rise of steam pressure during the night. Now, without a recording pressure gauge of some kind I do not see how anybody could tell in the morning to what point pressure might have risen during the night, although in the morning it might be even lower than the regulation limit for the night time. Of course, with a record made by a gauge during the night, locked up in

the superintendent's or the manager's office, and out of reach of the fireman, the unexpected and perhaps dangerous pressure to which a boiler might be subjected can be known and precautions taken accordingly.

In discussing this class of instruments before this Society, it is of course imperative that we be entirely impartial as between manufacturers of the different instruments, but, nevertheless, such a position should not deter an engineer from stating his convictions.

There is a feature which I have noticed in recording gauges, and that is the evident vibration of the pencil or marker in delineating the record of pressure.

I would like to ask Mr. Bristol how much force is applied in these connected tubes which he shows in his gauge. My idea is that with considerable pressure one way and considerable resistance the other way, the friction of the machine itself entirely disappears and we get a steady line, which of course is the important object sought. As far as I can see on Mr. Bristol's chart, the record seems to be very clear.

Mr. Jarvis B. Edson.—I am very much interested in this instrument, because there are many places where increased "travel" directly from the pressure itself is very necessary, and of the methods now in use none give it except to a limited extent. Still, without wishing to criticise the construction which in this new form gives the increased ratio of travel, I would like to ask how we are to get along with the sensitiveness of the spring to external jars and vibrations due to the fact of its being supported at one end only. You will remember that the old Bourdon spring was faulty in this direction, and gave rise to Lane's improvement, consisting in closing the tube at both ends and opening it at the middle, from which point it was supported. This secured greater rigidity and adapted the gauge to locomotive use. Perhaps the author can explain what is to hinder the marker from making a false record due to these external oscillations. Another thing occurs to me to which it is well to call attention in this connection, although not in any way meaning to reflect upon this particular form of instrument. It is this, that it is one thing to see a record on a piece of paper, by whatever device made, but quite another to know whether it is correct. The difficulty lies in the fact that the instrumentalities we can employ for giving the travel or original indications

are limited in number, and none of them act fully up to what we would desire. Coupled with inaccuracies due to various causes, they also lack resilience or ability to respond promptly when the change in pressure or temperature occurs. If a mercurial thermometer is placed in heated water the mercury will rise rapidly at first and then appear to come to a point of rest, while if carefully examined it may be found still rising, until after exhausting much patience, we conclude finally upon the temperature indicated. Then if removed from the heated water to the external air, much time will be required for the mercury to settle to a point indicating the air temperature surrounding it. Meanwhile, if read, nothing but error can result. Now, it is just as essential that a heat or pressure measuring device, whether of the indicating or recording kind, shall be capable of answering promptly and accurately, as that it shall simply have large "travel." I have noticed in the application of this sinuous tube device to the indicating of temperature that it, too, did not respond readily. In the course of my business as manufacturer of pressure recording gauges, I am frequently called upon for a recorder of temperatures for refrigerating cars, warehouses, &c., and I have invariably to say that I know of no means whereby satisfactory results can be obtained which shall respond quickly and accurately and also be applicable to the many places where such devices would be acceptable. In a vulcanizing boiler of a rubber factory, for instance, the temperature is to be maintained at different points during different periods of the night, but in the morning when the superintendent finds the valuable contents ruined he is unable to prove through what range the watchman has allowed the temperature to vary. Then, again, to be useful as a recording thermometer, it must be capable of being placed right in the medium whose temperature it is desired to record. You can't accomplish anything by placing the one shown us in the steam-jacket of a vulcanizing boiler; and if you don't so place it you won't get at the facts, even admitting that it possessed the properties which I have enumerated. So that, after all, what we want in these directions are sensitiveness, both up and down, and adaptability. If a recording device is sensitive without power, it cannot record the whole truth, and therefore is not adapted to the purpose indicated. To illustrate: Suppose the marking pencil or device causes friction sufficient to restrict the full throw of the carrier or pointer when

the pressure is increasing, it is obvious that the record will come short of the extreme height of pressure or temperature, as the case may be. Then when the latter is decreasing, the restricted travel due to such friction will prevent our knowing how low the temperature or pressure has fallen, and, in other words, we shall never know the whole truth, although that is what prompts the employment of these devices. Quick resilience, then, and power to overcome friction are indispensible in a recording device, and I arrived at the conclusion some eighteen years ago that there was only one way, and that was to increase the surface against which the pressure acts, and to resist it with proper metal, properly tempered to make a measuring spring of it. I thought steel was the best. While the instrument seems very nice working with a fluid marker, it appears to me that it should be so fortified as to enable it to mark with a lead point; because we want records of longer periods than a day, and the device should have power to overcome the friction necessary to wear off the pencil point. In connection with the subject I am very glad to know that somebody else is going to call the attention of the practical people of the country, as well as the steam users, to the advantages to be derived by recording the steam pressure. On water works systems, of course, there is a large need of them, as everything depends upon their pressure; but with steam in particular I know that a vast deal of data and risk go unknown. Facts are constantly coming to my attention growing out of the employment of steam recorders which show that their use should be imperative. In a building on Broadway, costing three-quarters of a million, fires were recently banked for the night as usual, but the wind changed both in direction and velocity after the fireman had gone, and the result was that the steam roared for an hour or two in the middle of the night to the alarm of those in the neighborhood. The safety-valve was operative in this case, as its name implies, and so prevented overpressure and an explosion, but the damper regulator closed when the pressure got up, and so shut in the heat of the fire and made the conditions all the worse. It may seem a trivial matter, but if the boiler had become short of water, due to steam generated, and blown off so unexpectedly, or if the safety-valve had been overloaded or stuck in its seat, as is often the case, an explosion would have resulted. The recording gauge record now shows the man where his steam

gets to during his absence, and he knows better how to guard against a repetition of such an occurrence. In another instance a recording gauge showed how a city boiler inspector tested a boiler by jumping the pressure for an instant to only five per cent. beyond what the boiler regularly carried and then releasing it, not even continuing the test long enough to pass around the boiler and learn how it looked under the testing strain. The record traced by the rising and falling pressure was a single vertical line, showing that the test was simply jumped on for an instant, and was more injurious and misleading than beneficial. I could cite, if time allowed, innumerable instances from actual occurrences to show how indispensible the pressure recorders are; but it takes much time and effort to convince the average steam user, or engineer, for that matter, that there is anything of value to be learned by their employment, and I gladly welcome this new candidate in the field of labor.

Mr. Alfred R. Wolff.—The point which the gentleman makes about the use of recording gauges I think has two aspects, and one is very favorable to the apparatus which has been show this afternoon. I think engineers as a class appreciate the value of recording gauges, and still they often find themselves at a loss to be able to apply them. It has been my fortune to specify for a great many steam plants in this city for important buildings, and still I find myself unable to include the recording gauge as often as I would like to. Although consulting engineers have the reputation that their employment means greater cost of plant, the funds placed at our disposal are such that, if we wish to make a decided success, there are so many things which are absolutely required, especially in liberality of capacity and proportion, that we cannot find the means, much as we should desire, to include recording gauges if they are expensive. Now, it appears to me that this Bristol gauge can be constructed very cheaply and put on the market cheaply, and if this is the case, even were its record not quite as exact as that of others, it would meet the great need. The greatest advantage of the recording gauge is to put a check on the engineer, and if that check is within one-half of one per cent., it will afford all practical means for the greatest use to which the gauge can be put; and while I have no doubt Mr. Bristol is prepared to point out that he gets scientifically accurate results, as I believe he does, I still maintain that if his records were not absolutely

scientifically accurate, and the apparatus can be furnished at a reasonable price, it will fill a very large need.

Mr. W. H. Bristol.—The inquiries of both Mr. Hague and Mr. Edson would indicate that they supposed the tube of the instrument would be quite sensitive to any jar to which it might be subjected, but this is not the case, for the tube is only about six inches in length and is considerably stiffened by the strip secured along the bends. It must also be noticed that the inking pointer is attached to the moveable or free end of the tube, and that the pen rests against the revolving chart for receiving the record. The tube is thus practically supported at both ends and is therefore quite rigid. For special cases where extra rigidity is required, a pair of parallel tubes could be employed by connecting their moveable ends. Such an arrangement would be quite analogous to Lane's improvement of the ordinary Bourdon gauge, of which Mr. Edson spoke, by which it was made available for locomotive use.

Mr. Hague asked about the force obtained by the motion of the tube. I would reply that this may be varied at will by changing the proportions of the tube. In the instrument exhibited it is $\frac{5}{8}$ inch in width. When the pressure is applied the tube responds instantly with considerable force, much more than is necessary to overcome the friction of the inking point upon the chart.

As the chart is continually moving, actuated positively by the clock work, in a direction at right angles to the motion of the recording pen, it is evident that when a change of pressure occurs the pen will at once readily accommodate itself to the true position due the change, thus giving accurate records.

I have found no difficulty in obtaining good results by a brass point and the so-called metallic paper, which is sometimes employed for indicator diagrams, and have no doubt that a lead pencil could be used if it was desired. For my part, I prefer using ink, as it makes a distinct and clearly defined record which is not so liable to become erased as a pencil line.

In exhibiting the pressure tube as applied to a thermometer, it was wet from an ice-water bath, and was so arranged that I was unable to apply the heat of the flame to advantage, hence it did not respond quickly; however, had I a vessel of warm water into which the tube could have been placed it would have

acquired the temperature of the water at once, and consequently have given a corresponding deflection.

In the application of the pressure gauge as a thermometer, its sensitiveness will, as in any thermometer, be limited by the facility of the tube and its contents to acquire with promptness the temperature of the surrounding medium.

As to the gauge's being scientifically accurate, I would say that I believe it to be as nearly so as it is possible to graduate it in comparison with a standard test gauge.

Some of the instruments have been in continual operation for from one to one and a half years, and I find that the tubes do not change or lose their elasticity.

Mr. Wolff called attention to the fact that the gauge was simple, could be easily constructed, and, consequently, placed upon the market at a moderate price—one which would make it possible for engineers to include a recording gauge in their specifications. This point is, I believe, a very valuable feature of the instrument shown, and will probably make recording gauges a necessity in many places where heretofore they have been considered a luxury.

The gentlemen have very clearly set forth the great importance of recording gauges, and it is very gratifying to hear directly from experts in steam and hydraulic engineering.

CCLXIX.

HOW TO USE STEAM EXPANSIVELY IN DIRECT-ACTING STEAM-PUMPS.

BY J. F. HOLLOWAY, NEW YORK CITY.
(Member of the Society and Past President.)

In offering the paper herewith presented to the American Society of Mechanical Engineers, it seems but fair to you, and just to the author, to give at least some of the reasons which have influenced him in the selection of the subject to which your attention will be briefly called.

In the first place, the peculiar mechanism described in the paper is the continuation, if not the culmination, of an invention distinctly American—an invention which, from its first conception nearly fifty years ago to within a recent date, was, through all its transition stages—from a little apparatus at first designed to supply feed water to a small boiler on a canal-boat, up to the immense engines now used to supply water for the largest cities of our own as well as foreign countries--the outgrowth of the study and patient industry of one who, in recognition of his great interest and invaluable aid in the formation of this society, has by common consent been named on the foremost page of its history as one of its "Honorary Members in Perpetuity--Deceased Founder of the Society."

If, in what will be said of the invention of the direct-acting steam-pump, and of the various subsequent improvements thereon, there shall seem to be a direct or implied tribute of respect to the memory of Henry R. Worthington, I am certain that such a tribute will awaken a hearty response in the minds and memories of all the members of that small group, who a few years ago came together in New York City, and founded a society whose growth in membership has not only been most marvellous, but which, better than all else, has come to take its place among kindred societies as the acknowledged leader of mechanical ability and achievement.

The special contrivance to which I shall later on call your

attention, and which, amplified and perfected since the death of Mr. Worthington, has been added to the direct-acting steam-pump, and which by its action has lifted that class of machinery to the higher, if not the highest, realms of economic duty, has been the result of the study, experiment, and ingenuity of a member of this society. I speak of this fact at this time to illustrate, to some extent at least, the truth of the saying, that an inventor, like a prophet, is not without honor save in his own country. That it was left to foreign engineers first to discover the value of these later inventions, and to adopt them, and for the judges of a foreign exposition to award to their performance the highest reward in their gift to bestow—the grand prize—is in the main due to the fact that so quiet has been the introduction of these new and important features in pumping machinery in this country, so modestly have their merits been announced, that few engineers are aware of their existence, and fewer still have had an opportunity to study the principles upon which they are founded, or the results which they produce.

Fortunately, to this audience it is unnecessary to recount the history of pumping machinery, or to enlarge at length upon the difficulties as well as the dangers to be overcome in the conveyance of water through long lines of pipes. About me on all sides are veteran engineers, who, in the holds of steamers, the sump of the mine, or in the well of the pump-shaft, have wrestled with split pipes and blown-out joints, improvising clamps for the one, whittling wooden wedges for the other, and all on account of that aquatic beast never yet seen but often felt and heard, and always feared—the hydraulic ram.

All will, I think, agree that the perfection of pumping machinery would be an arrangement of mechanism which would be self-contained, easily reached in all its parts, inexpensive as to its own cost, and also as to the foundation upon which it rests, which would possess a self-restraining power which would become automatic in case of the breaking of mains or wide variations of loads —a machine which, while it possessed ample power to force the water through the pipes at all times, would at the same time do so with an elastic power which, while yielding to sudden and remittent resistance of the water in the mains, would still steadily keep it on its onward movement. If such a pumping-machine can be had, in which one or more steam cylinders can be directly attached to the pump-plungers, and without the use of cumbersome vibra-

ting beams, or heavily rimmed revolving fly-wheels, together with the required intermediate connections of shafts, cranks, connecting-rods, and complicated valve gear, etc., we shall have greatly simplified the pumping engine as formerly built to supply cities with water.

Such pumping engines we have in very extensive use, and their value is well attested by the fact that a large number of firms are engaged in their manufacture. Valuable as are these engines, numerous as are their good points, improved and perfected as they have been, of late years, by the compounding of their steam-cylinders, they have always and still lack one thing—one thing which their most ardent admirers have been obliged to confess as not only desirable, but imperatively necessary, before they can enter the list among the most economic engines for raising water. That one thing is the ability to cut off the steam within the cylinders, and to complete their stroke by the expansion of the steam remaining in the cylinder. Of late years the economy of direct-acting pumping engines has been somewhat increased by compounding the steam cylinders, and possibly some further advantage may be derived from the use of triple-expansion cylinders; but the complication of parts, and especially the inaccessibility of pistons and packings, would seem to limit progress in that direction. Having shown what the direct steam pumping-engine, valuable as it is in many respects, still lacks in order to produce the most economic results, we are now ready to proceed to describe a most ingenious device by which this long-felt want has been supplied.

The ideal card taken from the water cylinder would be a card square at each end, and in which the sides would be parallel to each other. That such cards are rare is true, but that they are possible is also true.

Now, in order to use steam expansively in one cylinder, it must be admitted at the commencement of the stroke at a pressure higher than that required to perform the average work assigned it. Just how much higher will depend on the ratio of expansion sought to be obtained. Yet, if steam under such conditions should be admitted to a pumping engine of the kind under consideration, it is easy to see that our ideal water card would be at once distorted beyond recognition; while, if we cut off the steam during a portion of the stroke, its pressure, by reason of its expansion, would fall below a point necessary to propel the plunger, and the

pump would stop short of the end of its stroke. What is wanted is some means by which the excess of power due to high steam, admitted at the commencement of the stroke, can be stored up so as to be available toward the end of the same stroke, and thus equalize the power to the load; so that the propulsion of the plunger shall be equal and uniform throughout the entire stroke, and so that the water column be so steady in its flow as to show no perceptible pulsations.

The cut, Fig. 16, shows a sectional elevation of a compound direct-acting steam-pump, having attached to it what has been called the high duty attachment.

To ordinary compound direct-acting steam pumps, as usually built, there is attached a plunger-rod which projects through the outer end of the pump chamber, and around which there is the usual stuffing-box for packing the same. On the end of this plunger-rod is fastened a cross-head, which moves in guides which are bolted on the outer end of the pump. On this cross-head and opposite to each other are semi-circular recesses. On the guide plates are cast two journal-boxes, one above and one below the plunger-rod, both equidistant from it, and at a point equal to the half stroke of the cross-head. In these journal-boxes are hung two short cylinders on trunnions, which permit the cylinders to swing backward and forward in unison with the plunger-rod. Within these swinging cylinders are plungers, or rams, which pass through a stuffing-box on the end of the cylinder, and on their outer ends they have a rounded projection which fits in the semi-circular recesses in the cross-head; and consequently, as the cross-head moves back and forward, it carries with it these two plungers, which in turn tilt the cylinders back and forward on their trunnions. These swinging cylinders are called "compensating cylinders," and they are filled with the fluid being pumped.

The pressure on the plungers within the compensating cylinders is produced by connecting these cylinders through their hollow trunnions with an accumulator, the ram of which is free to move up and down as the plungers of the compensating cylinders move in and out. The accumulator used is of the differential type; it has below a small cylinder filled with water or oil, within which its plunger moves, while above it has a larger cylinder filled with air, and within which there is a piston-head which fits closely to the cylinder, and is at the same time attached to the top of the plunger in the lower cylinder.

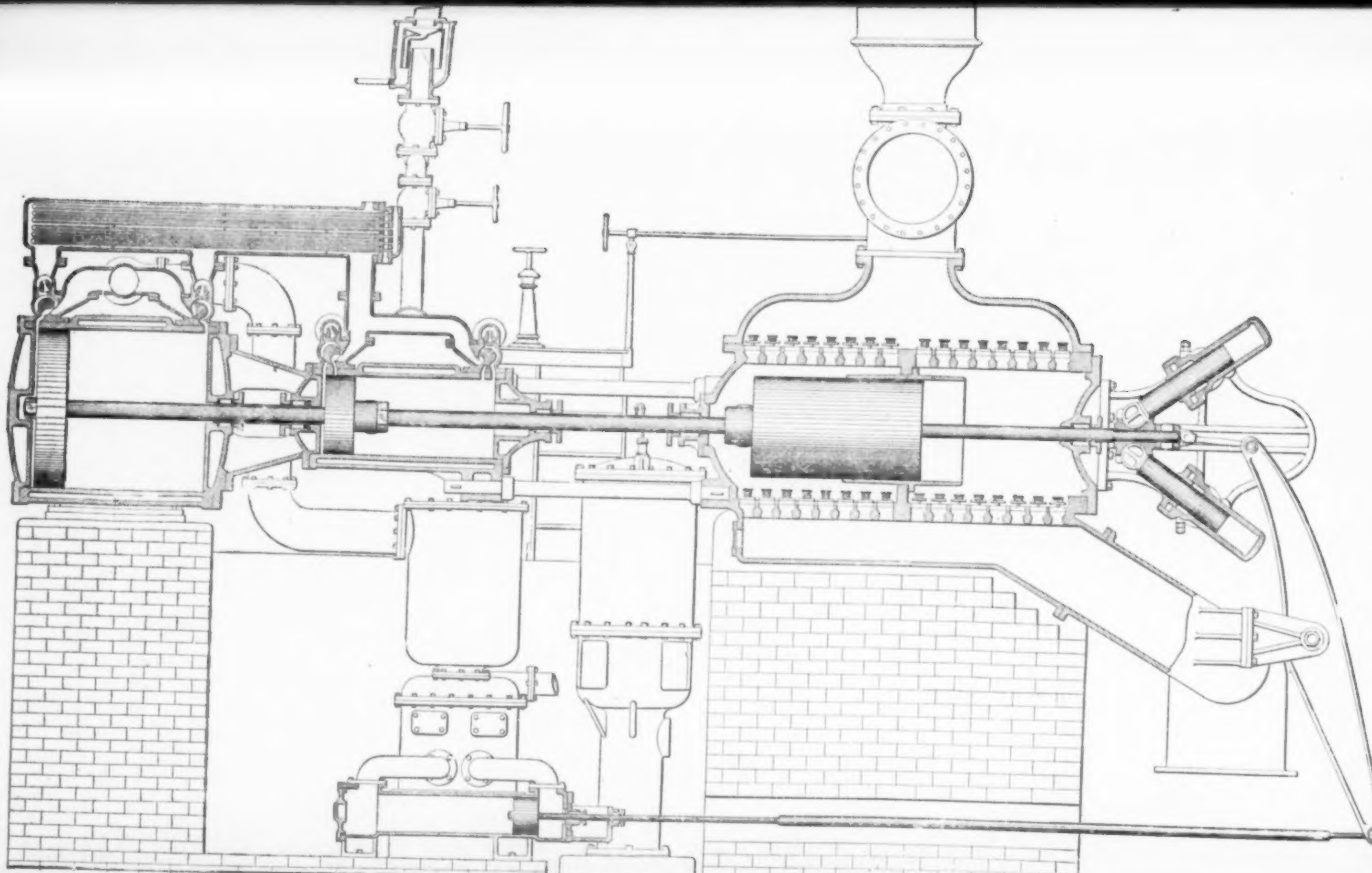

FIG. 16.

By this arrangement it will be seen that the pressure per square inch on the plunger or ram of the accumulator will be the pressure per square inch on the piston-head in the upper cylinder, multiplied by the difference between the area of the piston-head and the lower plunger. This difference of areas is a matter of calculation, based upon the particular service for which the pump is constructed. The pressure in the air cylinder is controlled by the pressure in the main delivery pipe of the pump, as it is connected to that pipe. This connection with the main has another very important use, as the power exerted by the compensating cylinders is a very considerable part of the power used in driving the pump-plunger at the latter part of its stroke, and it will be seen that if, for any cause, either by the breaking of the main or otherwise, the load is entirely thrown off the pump, the plunger cannot make a disastrous plunge forward, for the reason that the steam in the steam cylinder is, by reason of its expansion, too low in pressure to drive it, while the fall of pressure in the main has robbed the accumulating cylinders of their power.

Having briefly described the construction of this novel attachment for the purpose of using steam more expansively than ever before in direct-acting steam-pumps, we will examine, with the aid of a diagram, as to the varying forces, both retarding as well as advancing, produced by the action of the compensating cylinders. In Fig. 17 we have, above and below the base line, *A B*, a curved line which graphically illustrates the amount of power exerted, and how it is divided during one stroke of the pump, *A* being the beginning of the stroke, f'' the half-stroke, and *B* the end of the stroke. In the illustration, under the curved line we see the position of the compensating cylinders at various parts of the stroke, and from which position the line of curvature above is produced. At the commencement of the stroke at *A*, it will take an amount of power equal to the line $a''l''$ to push the plunger in the cylinder, and against the load on the accumulator; as the plungers are driven in, the angle of inclination of the plunger to the centre line of the pump-plunger changes, so that at each ordinate it takes less and less power to push them in, and, as a consequence, the retarding effect on the pump-plunger is less and less, until it arrives at the half-stroke, when the plungers stand at right angles with the pump plunger-rod, and they exert no effort to drive it in either direction. As the pump-plunger begins to pass its half-stroke, the plungers of the compensating

cylinders begin to push outwardly, and begin to exert an influence to propel the pump-plunger forward toward the end of the stroke;

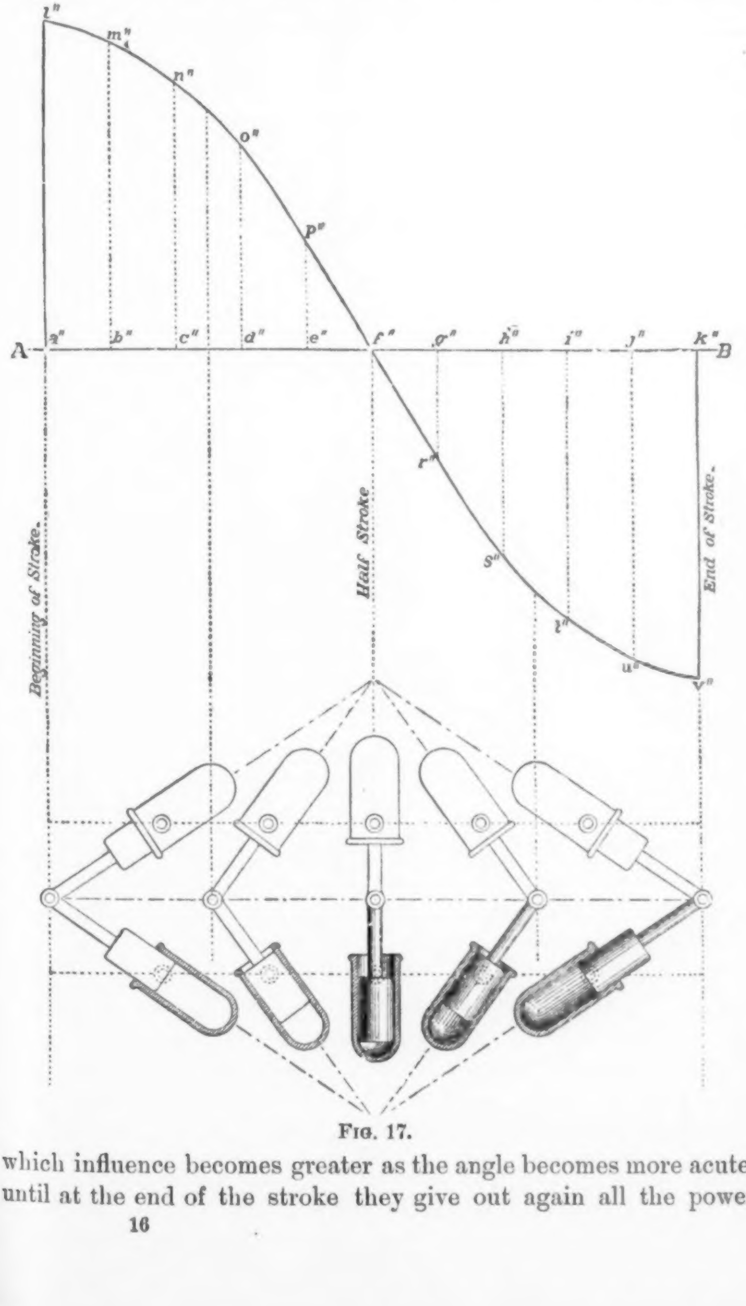

Fig. 17.

which influence becomes greater as the angle becomes more acute, until at the end of the stroke they give out again all the power

they took up at the beginning. This line of resistance and impulse can be varied by changing the points of suspension of the compensating cylinders, or the load on the plunger of the accumulator, so as to suit different steam pressures and various points of cut-off. In Fig. 25 we have a card taken from the water end of the same pumping-engine as the steam cards were taken from, *XX* showing the line of atmospheric pressure, and *Y Y* the zero or no-pressure line. In this water card the ends are vertical, and the top and bottom lines are practically parallel, and in these respects it fulfils all the requirements of our ideal water card.

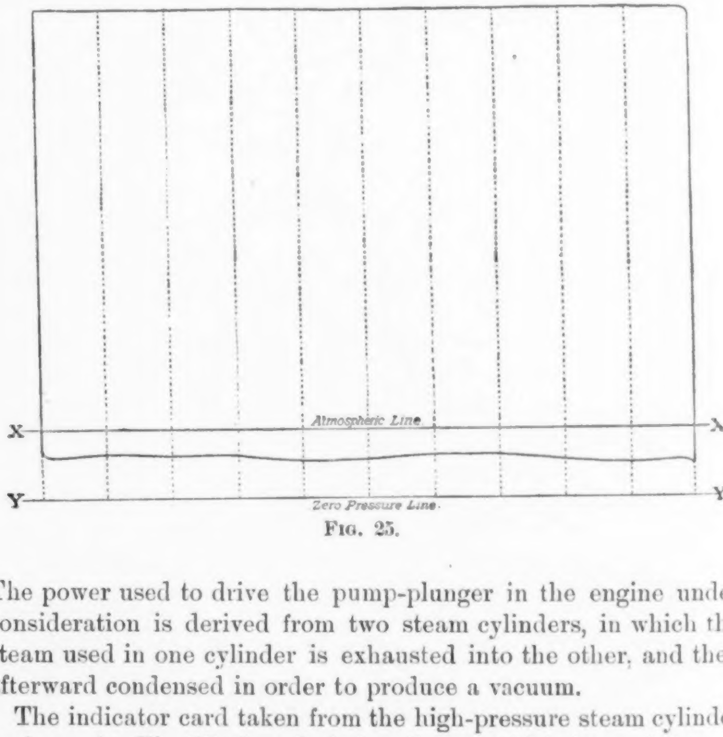

FIG. 25.

The power used to drive the pump-plunger in the engine under consideration is derived from two steam cylinders, in which the steam used in one cylinder is exhausted into the other, and then afterward condensed in order to produce a vacuum.

The indicator card taken from the high-pressure steam cylinder is shown by Fig. 18, the admission line of which is straight and perpendicular to the line of motion; this is due to the fact that in pumping-engines of the kind which we are describing there is a slight pause at the end of each stroke, which not only allows the pump-valves to seat themselves quietly, but it as well fills up the clearance and steam ports to the full pressure before the piston starts. In this diagram *XX* represents the line of the atmos-

pheric pressure, and *Y Y* the line of zero or no pressure. The power exerted in this cylinder up to the point of cut-off and from that to the end of the stroke is shown by the line *l*, *m*, *n*, *o*, *p*, *q*, *r*, *s*, *t*, *u*, *v*, which is the steam line of one stroke of the high pressure piston. The return stroke is shown by the exhaust and compression line, *k*, *j*, *i*, *h*, *g*, *f*, *e*, *d*, *c*, *b*, *a*.

The actual power exerted in this cylinder will be the pressure above the line *Y Y* at each of the ordinates, less the back pressure at the same ordinate. If we take this actual pressure, and apply it to the same number of ordinates, all of which shall start

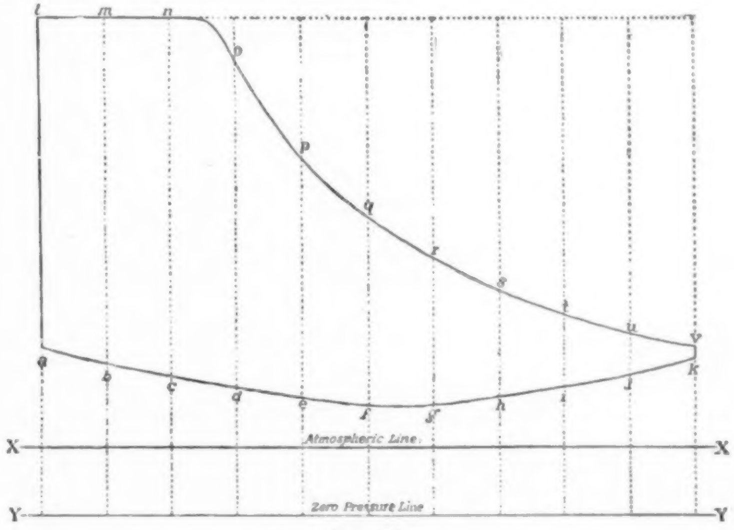

FIG. 18.

from one common base line, as at *A K*, Fig. 19, we will have a curved line, as shown by *A′*, *B′*, *C′*, *D′*, *E′*, *F′*, *G′*, *H′*, *I′*, *J′*, *K′*, and which will be the available steam pressure, or power, at each part of the stroke of this piston. In Fig. 20 we obtain by the same process the line of pressure in the low-pressure or expanding steam cylinder, and on the same number of ordinates, and from each of which we must, as before, deduct the back pressure above the line *Y Y*. When we have done so we will have, as before, a curved line, showing the available pressure in the low-pressure cylinder. This curved line of power or pressure will be seen in Fig. 21.

As, in this case, the low-pressure cylinder has four times the

area of the high, we multiply the pressure shown in Fig. 21 by four, and this result, as shown by Fig. 22, is the power exerted on the pump-plunger by the low-pressure piston. Having

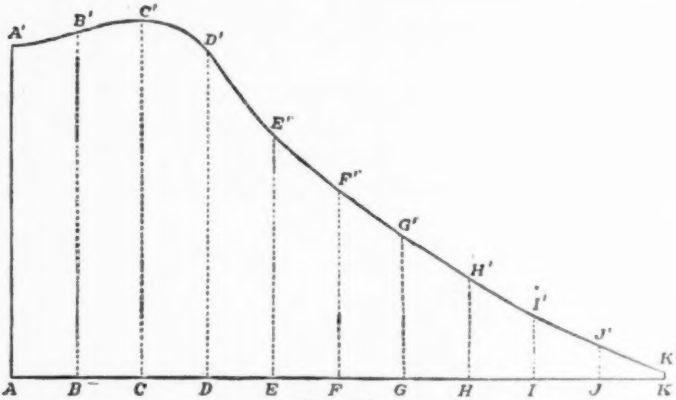

FIG. 19.

now the power developed on the H. P. and L. P. cylinder shown separately, in order to know what power they exert when combined we add one to the other, which gives us the card as shown

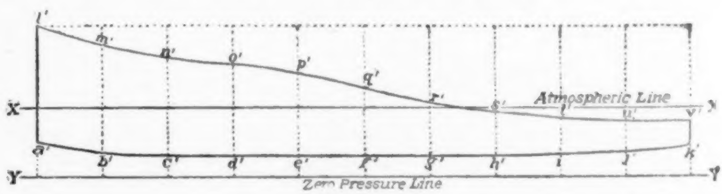

FIG. 20.

by Fig. 22, which not only shows the total power exerted by both pistons, but also shows just how much power is exerted at every part of the stroke. An examination of the curved line of pressure

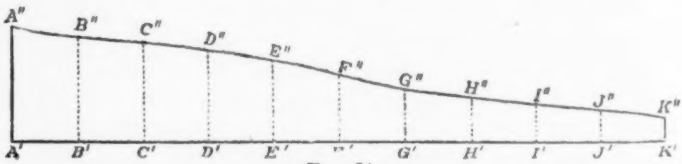

FIG. 21.

or power on this card (Fig. 23) will clearly show how unsuitable such a power is to produce a uniform steady motion on a water column when connected directly to the plunger of a pump. In

Fig. 24 we have a composite card in which all the previous steam, water, and compensating cylinder cards are combined in one card.

First.—Within the lines *A*, *W*, *W'*, *K*, we have the water card previously shown in Fig. 9.

Secondly.—The space enclosed within the lines *A*, *W*, *A"*, *B"* *C"*, *D"*, *E"*, *F"*, *G"*, *H"*, *I"*, *J"*, *K"*, and *K*, *J*, *I*, *H*, *G*, *F*, *E*, *D*, *C*, *B*, is the steam cards Fig. 19 and Fig. 22, all combined in one card, as in Fig. 23, the top curved line of which shows not only the total steam power used during one stroke of the piston, but it also shows just how much power is exerted at each ordinate, which in all these cards enclose one-tenth of the stroke.

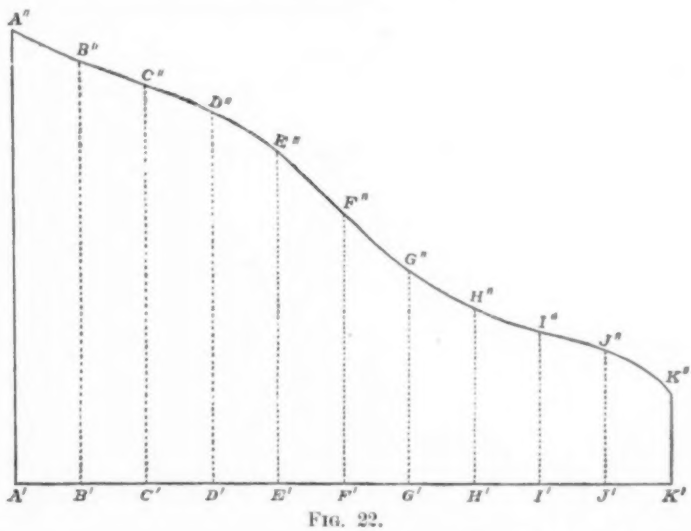

FIG. 22.

Thirdly.—In the lower curved line, and which crosses the base line, *A K*, at *F*, we have the line of effect produced by the compensating cylinders, as shown and explained in Fig. 17. The end sought to be accomplished, it will be remembered, is the use of steam at a high pressure in the steam cylinders, cutting it off during a portion of the stroke, and at the same time to have the power exerted by the steam on the water column, and through the movement of the pump-plunger, exactly equal to the resistance of the water, when moving under a perfectly uniform pressure. In other words, what is wanted is to have the very irregular curve of the steam pressure brought down to the parallel lines of the water pressure. As it will readily be seen by an examination

of Fig. 24, the power exerted by the steam at the beginning of the stroke, and up to the middle of the same stroke, is largely in excess of the power required to move the water, as shown by the lines of the water card; while during the last half of the stroke it falls below what is required to move the water column.

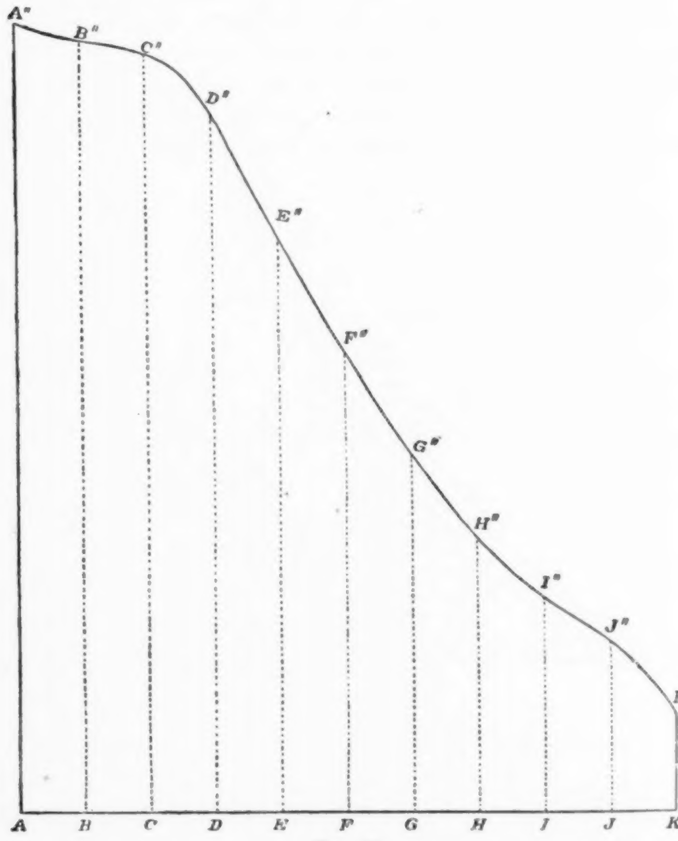

FIG. 23.

Now, if we examine the effect produced by the action of the compensating cylinders, we will see in what a remarkable manner they operate to correct the irregularities of the steam card.

At the commencement of the stroke at the left hand side of the card, Fig. 24, it will be seen that the power as shown within the lines W, A'', B'', C'', D'', E'', F'', is an excess of power—more than is wanted to move the water; but if you look at the lower left hand

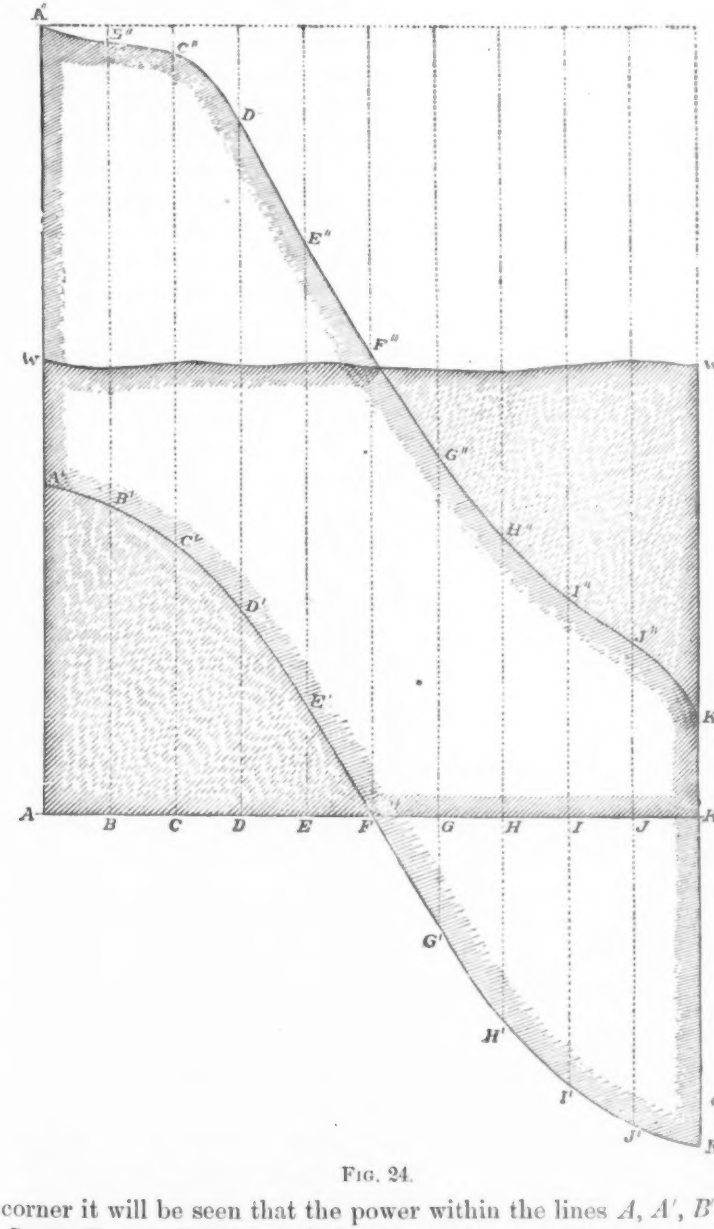

FIG. 24.

corner it will be seen that the power within the lines A, A', B', C', D', E', F', and F, E, D, C, B, has been taken out of the steam card and absorbed by the compensating cylinders; while on the right

hand side of the card, and which from F'' is the last half of the stroke, when the line of power in the steam card falls below what is required to move the water column, the compensating cylinders give back the power they had previously absorbed, and thus supply just the needed power to complete the stroke.

From this explanation of Fig. 24 it will be seen that the actual available power for the propulsion of the water, irrespective of whether the pressure of the steam is high or low, is the power that lies between the upper and lower curved lines, and which, during the entire stroke, is practically a uniform power.

As a proof of the remarkable manner in which the compensating cylinders do compensate for the irregular pressure of the steam as thus used, if you will take a measurement between the upper, or steam curved line, and the lower, or compensating curved line, on any of the ordinates, in any part of the stroke, you will find that the distance between these lines, and which is the measure of the available power for driving the pump-plunger, is exactly the same as is the distance between the upper and lower lines of the water card on any ordinate, in any part of the stroke of the pump. In fact, it may be said that the lines of power, as well as the lines of resistance, are practically parallel, and thus it is that there is exerted at all times just power enough, and no more, to force the water column along so quietly, so steadily, that on engines of this construction it has been found there is no possible use for an air-chamber on the delivery mains; and thus is the apparently paradoxical problem solved, of producing a perfectly uniform motion and pressure in a steam-pump using steam power variable to the last degree, and without the use of shafts, cranks, fly-wheels, or heavy vibrating beams.

There are many ingenious devices connected with the compensating cylinders and the accumulator, which, while interesting, it has not been thought best to occupy your time in describing, as they are only accessories to the grand result.

In closing, I avail myself of the opportunity of giving still another reason for the presentation of this paper at this time. Believing, as I do, that the invention described opens up a new and important era in the construction of pumping machinery, and that a careful study of the principles upon which it operates, and the manner in which it is constructed, as well as the results produced by its use, will confirm in your minds the favorable verdict already given concerning it by eminent engineers abroad, I felt

what I trust you will all deem a commendable pride in desiring that its first presentation and discussion, before an engineering society, should be before the society which has in so marked a manner honored the memory of the man whose name is so closely identified with the origin and success of the direct-acting steam-pump, and who, by his life and his life's work, did so much to elevate to the proud position it now holds, at home and abroad, the name—American Engineer.

DISCUSSION.

Mr. H. H. Suplee.—To refer to a piece of ancient history, I should like to call the attention of the members, in connection with this matter of compensating the inequalities of the power at different parts of the stroke, to the fact that many years ago, I think while James Watt was still living, there was a device attached to one of his beam engines when they were first applied to rotary purposes, which, in a very imperfect manner, suggests the germ of this device. There was a small cylinder which was open to the atmosphere at one end and entirely closed at the other. This was geared to the shaft of the main engine by means of suitable linkage—two to one, so that it should make two strokes, while the main cylinder made one. When the steam was first admitted to the main cylinder it had to lift this small cylinder against the pressure of the atmosphere. When the main piston had reached half stroke, the small piston had made full stroke. While the main piston was completing its stroke, the small piston was going back again. The resistance of the atmosphere opposed the force of the steam at the beginning of the stroke and assisted the force of the steam at the end of the stroke. This did not compensate nearly as uniformly as the cylinders of the Worthington pumping engine, but it did enable the rotative engines then in use to make much more uniform speed. I think this device was invented by Mr. Buckle, who was connected with the Watt establishment, and it will be found described in Mr. Bourne's large work upon the steam engine. It seems to have lain entirely dormant until it is now reapplied in a much improved form for pumping engines.

Prof. J. B. Webb.—The device employed in this improved pump stores up the power, so as to allow the steam piston to move uniformly while the water piston moves uniformly. I

should like to ask Mr. Holloway, however, what objection there is to disconnecting the pistons from the same continuous piston-rod and connecting them by mechanism in such a way that, while the water piston moved uniformly through its stroke, the steam piston would move with an accelerated motion during the expansion. With such an accelerated motion of the steam piston as would make the work developed by the steam the same for each fraction of a second, *i.e.*, constantly equal to that required for pumping the water, and with a suitable connecting mechanism, adapted equally to the forward and return strokes, there would be no necessity for a storage of energy.

Prof. J. E. Denton.—It is a matter of great pride to all American engineers to contemplate the principles and the success of the Worthington pumping engine in the form in which it was developed by our late distinguished member, Mr. H. R. Worthington. The improvement now under discussion, I understand, provides a means by which a direct-acting pump, with no crank or fly-wheel, may expand steam at least ten times with perfect success.

In a general way I have understood that there was a compensating action in the new attachment, but I never appreciated before how perfect the adjustment is, by means of the varying obliquity of these plungers acting against the constant pressure of the accumulators.

Prof. R. H. Thurston.—It is a great pleasure to be able to say that, among all the devices which were presented to the Juries at Paris last summer, there was no one, so far as my own knowledge and hearing went, which attracted more attention than this device of Mr. Worthington's. It was looked upon as a most beautiful embodiment of American ingenuity. I do not know which was more commented on: the perfection with which the apparatus worked or the beautiful simplicity of the device itself. I think it was its simplicity quite as much as its effectiveness which secured for it the grand prize—a prize which is very rarely accorded to any invention. The members of the Jury, and engineers who were also interested in the matter, were very desirous of securing more extended accounts of the performance of the engine as thus improved, and we could only refer them to the results of a trial by Professor Unwin, which was published in London Engineering—and I think republished by the builders of the engine—the result being to show that the engine has an

extraordinarily high economy, and that it is thus brought into the highest rank, beside those pumping engines which have hitherto held their own distinctively on the score of economy.

Mr. Chas. E. Emery.—It has been my pleasure for years to note the development of the Worthington pumping engine. The award at the Centennial Exhibition was written by me, but at my suggestion signed by one of the Judges from Great Britain, and I am happy to say was very pleasantly received by Mr. Henry R. Worthington, of whom Mr. Holloway has properly spoken in so pleasant a manner. In that award it was stated that the effort during the stroke of one of the two cylinders of a duplex engine blended into that of the other, so that although there was reciprocating motion, the water column moved on uninterruptedly and almost without change of velocity. This new invention—due principally to Mr. Charles Worthington, though doubtless under discussion before the death of the father—maintains the same uniformity of action, the same blending of the force exerted by one of the duplex pumps into that of the other, and maintains practically a uniform velocity, although the steam pressure in the cylinder varies greatly on account of the high degree of expansion. This latter improvement I may say rounds out, develops, and completes the original conception, and secures in practice results so satisfactory in every respect that nothing further can be desired.

Mr. Chas. A. Hague.—The question as to the relative economy between the older form of the Worthington engine and this new type has been incidentally brought up, and, having some general information on that point, I would like to state it. The records are published and are accessible, and in general terms the economy is about one-third. It saves about one-third of the fuel over the older form of the Worthington engine, principally, of course, because the ratio of expansion is increased and the steam is cut off and expanded in both cylinders.

One of the records to which I might refer is in connection with pumping oil with a horse-power of 500 to 700 against an oil pressure of from 800 to 1,000 pounds. The difference in twenty-four hours in fuel was the difference between 23,000 pounds of coal and 35,000 pounds of coal. The conditions are exactly the same, and the engines are side by side. The cylinders in the high-duty engine are two 41-inch high-pressure cylinders, with two 82-inch low-pressure cylinders, all of 36 inches

stroke, and taking steam up to one-quarter stroke at 100 pounds gauge pressure. The engine alongside of this new one has two steam cylinders, each of 33 inches diameter high pressure, two low-pressure cylinders 58 inches diameter, all of 36 inches stroke, taking a lower pressure steam and the steam following full stroke, the only expansion being that due to the relative areas of the pistons and the loss of the space between cylinders.

Mr. J. F. Holloway.—As to the question of economical performance, that is one which, of course, is a matter of importance in the engineering sense of the word. It is a matter which, however, must be made up by the records from time to time, and I have only to say that it is an indication of the very thing which I have said in the paper—that the people who build the engine have not flooded the society with figures and pamphlets. I did think that this mode of producing a result that has long been sought for was one of very great interest, and I am quite sure it is, from the remarks that have just been made. I simply brought it forward as an engineering problem, or rather the means of solving a problem which all of us have stumbled over since the introduction of direct-acting steam-pumps. We have all recognized the value of direct-acting pumps in many ways, as has been fully shown by the fact that they are so largely used in this country, and so largely used abroad. But, as I have said, we have always conceded that they are an expensive pump to run, so far as the use of steam is concerned, and there has been great difficulty in devising plans by which that expense might be decreased, so as to add increased value to that character of pumping machinery.

I am very glad to hear the matter spoken of by Mr. Suplee, in regard to the device which Watt had thought over and planned out. I never heard of it before, and he has himself explained that it is not just like this. But that James Watt, whom we all look up to as the father of the steam engine, among all the many and various things which he thought of, and which he worked out, and which he did the best he could to perfect, had this in mind, gives me a still greater reverence for that great man whose name stands at the head of our profession, and to whom we all look up with respect.

Mr. Webb spoke of other devices for storing up power. He wants to know if there may not be still others. If he can tell me what mechanical engineers will not do some day or other, I

will tell him what may be done in this respect; that some other way may be devised, I doubt not. The world is open for the best thoughts and the best intellects, and engineers are the people to whom it looks in order to bring both out; and, while I cannot see what is in the future, I am quite sure that this is one of the subjects which will commend itself to the study and investigation of all.

The remarks by Professor Denton and by Professor Thurston are of great value, because they have both studied the matter more fully than the ordinary engineer does, and because it is a matter of interest to persons in their especial line, as well as it is with those who have occasion to use pumps. Professor Thurston saw the engine running in Paris, about which I should be very glad to give you a history if there was time. There are other points as to the methods in which the engines are constructed, which I know would be of very great interest, and I will only say that if the gentlemen who built and invented the pump, and who have sat up at night thinking about it, would only consent to put on record a portion, at least, of what they know about it, the American Society of Mechanical Engineers would be very much interested in reading it, and would be very much indebted to them for it.

CCLXX.

STREET CAR GEAR FOR MODERN SPEEDS—THE COMING SELF-PROPELLED CAR.

BY S. J. MCFARREN, MCKEESPORT, PA.

(Associate Member of the Society.)

STUDENTS of current industries, familiar with the beginnings of invention and enterprise, as well as with their business control and direction, lack faith in some of the popular superstitions.

They often meet, for instance, the manufacturer who is too busy in the *details* of his business to learn its *principles*—who has not *time* to read technical journals, or reports of society proceedings and experiments in his specialty. While perhaps *specifically* intelligent, he is in a general sense ignorant, and is the veriest slave to the moss-grown usage of his business, instead of holding the princely control and leadership thereof with which the reportorial obituaries will endow him.

This case is even more frequent in the mechanical management of some of our greatest concerns and industries than in that of their commercial and social or labor departments. The descendants of the youth who carried the grist to mill in one end of the bag, balanced by a stone in the other end,—"because Grandsire did so"—have increased to the point of overflowing from theology, law, and medicine into mechanics, with special tendency to railroading in some form. The most frequent expression of the master mechanic of one of our great transcontinental railroads, during a visit of inspection by the writer some years ago, was, "We adopted that device in the airly days, sorr, and we've nivver changed it!"

A like slavish adherence to precedent and deference to reputation seem to be co-operating with the ignorance too common among inventors and company officials—and perhaps with the natural conservatism of manufacturers,—to produce some street railway practice curiously similar, and causelessly parallel, to that of the larger railroads, in which the parallel rods and counter bal-

ances of the modern locomotive embody the ideal of mechanism. One of our leading electrical companies recently had for a motto, "No inventors need apply"; and an employee's suggestion for a much needed improvement was rewarded by a sarcastic rebuke and threat of discharge.

During the writer's residence in an interior Mexican State capital of some thirty thousand inhabitants, a native street railway company was organized and road built (3-ft. gauge). The cars came by wagon from the railroad station nearly two hundred miles distant, at a total cost for freight of about five cents *per pound.* They bore the mark of perhaps the best-known maker in the world, whose reputation seemed to have made unnecessary such old formalities as shop inspection before shipment. They had apparently been hastily altered from a wider gauge by the simple means of pressing the wheels closer together on the axles, which were of the "cold rolled" type, *without* moving in the sills or journal boxes. This method left some ten inches clear axles *outside* of wheel hub at each end and *between* that support and the bearing which was to carry the load—say three or four times in excess of good practice! The writer's suggestion, to cut off the axles and move the sills and journals in to correspond with the narrow gauge, was met with a child-like trust in the manufacturing "house" and the fear of *lessening the stability* of the cars by thus *narrowing their support!* The (Mexican) foreman went so far as to say that he had himself seen a railroad (Mexican Central) whose entire equipment was of this identical construction.

I need scarcely add that those axles "cranked" to a permanent set on the first ensuing holiday or that a sorry lot of wheeled imitations of crippled cattle are to this day sustaining the reputation of the "house" in that vicinity.

The science of the Mexican foreman is equalled in the recent advertisement in a great technical journal, descriptive of a device for adding to the longitudinal stability (decreasing the "rocking motion") of a street car *without* increasing its "wheel base" or distance between axles, which plan is mentioned only to be avoided by the genius of this inventor as impracticable! A brief inspection of the simplest diagram of the "angle of stability" would evidently be a revelation to this expert; and in view of their respective opportunities, the Mexican seems the best informed. Since the angle of stability must have for its apex the

centre of gravity of the load, and for its base the distance between the wheel contacts which furnish the ultimate supports of that load, it is not an exhausting effort of the intellect to perceive that all devices which fail to lower the center of gravity *or* increase the distance between wheel contacts (laterally or longitudinally, as the case may be) are useless.

The present increase in street railway construction has greatly increased the percentage of uninformed buyers, and made a harvest for all and every class of builders of street cars, who have with one accord exhausted their resources upon the car *bodies*, which are "seen of all men." Thus we find that the two best out of three leading street car *gears* are made by concerns which do not build *cars*. Now the gear *is* the car,—as the legs and feet are the *horse;* and its neglect by manufacturers is emphasized by the severe requirements of modern speeds and mileage,—doubling and trebling former practice as they do,—a daily mileage per car of one hundred and fifty miles and more, at a rate of ten to fifteen miles per hour being already common.

These points are especially marked in electric railway service where variations in speed are not only excessive compared with horses or cable, but the problem is often that of *self-propulsion.* The rigidity of old gear constructions, for instance, is almost prohibitory of "self propulsion." Those managers of electric "systems" whose worship of the subtle fluid did not entirely dwarf their estimate of the mechanical problems involved, have attained their brightest success by discriminating adaptation of devices long tried and proven on the larger railroads, and it is in this direction that we look for future progress.

Though not in the scope of this paper, the permanent way is so closely related to the rolling stock that improvement in the last presupposes change in the first; and it is noted in passing that general recognition is already given by equipping companies and their patrons to the need of heavier rails, of stiffer section, better alignment and surface, etc., also that the best practice dictates grooved rails on all curves and the elevation of the outer rail wherever allowed by grade of street. The utility of the last, for street-car speeds, was gravely questioned by a conservative engineer of an electrical company, but he has been unable to obtain any suspension of the forces of gravitation and momentum in support of his objection, or his derailed cars.

With a fit road-bed, the remedy for most of the discomforts due

to faulty rolling stock is so simple and easy that the public will demand it as fast as informed. Allowing that the usual diameter of wheels and height of springs are to remain unchanged, we have only to place the supports further apart (lengthen the wheel base) and the rocking or "gallop," on undulating track, also the swaying or "wag," on leaving or entering curves, disappear together if the gear is of the suspension or other flexible type. Without desire to provoke discussion from any one who holds to the impracticability of this plan, I note that for cars longer than 20 feet, probably the best way to reach this end is that already adopted by some cable and electric roads, of using two four-wheel pivoted trucks under each end of the car, a close imitation of railroad practice. For short cars up to 18 feet, however, two axles and four wheels will continue to be the maximum, and it is to this class which we are here limited. By the adaptation of the suspension or other systems long "standard" in railroad service, the wheel base of ordinary four-wheeled street cars may readily be increased 50 or 60 per cent. over present practice, with all the corresponding good results and *without* encountering even the smallest dragon of "impracticability" so much dreaded. Not only so, but traction on curves can thus be *lightened* instead of increased, so great is the resistance of the rectangular rigidity of old style gear. It is a fact that cars of 9 feet wheel base have been running successfully for 10 years on both horse and cable roads, on curves as sharp as 35 feet radius, and with unapproached economy of power and lubricant, and unexcelled comfort and ease of riding. The writer knows of one case where such trucks (four wheeled) are carrying 30 feet cars through curves of 45 feet radius, but considers this an evident case for 8 wheels, as it is safe to say that no car can ride easily with two or three times as much of its load and length *outside* of, as that *between* wheel supports. Indeed he would make the rule that *at least as much* of the car length must be *between* as that *outside* of supports, *i. e.*, only one-fourth of the entire length must be allowed to project at each end over and past the axles. Thus a better distribution of load on bearings would be secured, and neither axle would have the whole load to carry, with risk of breaking springs or wheels. A glance at the Thielsen or other suspension trucks of the ordinary passenger coach and many freight lines will show how much the suspension principle of construction must reduce flange wear and resistance. Other

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methods for securing flexibility are not wanting, but present limits do not permit description of them.

With sufficiently flexible gear and independent wheels it is believed that present wheel base practice (6 feet maximum) can be more than doubled and *lighter* draft on curves secured. Independent wheels need not be of the "loose" type, common in mines, etc., but may entirely avoid the many faults of that construction. There are many practicable forms of them which have stood severe tests in railroad service, and only failed of introduction by reason of devotion to standards and precedents and the exigencies of the "car-exchange" system and various other limitations from which street car service is and should be free.

More powerful brakes are necessary for the heavier and faster running cars of the power systems. The efficiency of the track brake has been demonstrated for years on the Pacific Coast, and only an attempt to dodge the patent office, by use of wooden instead of iron shoes, has delayed their adoption here. With them the greatest economy of wheels is secured. They should be applied by power, however. This is accomplished in a Western electric system by the use of compressed air maintained by a pump geared to the axles. The car momentum may be conveniently used for the same purpose.

From even this superficial consideration of the subject, its extent and importance are apparent with the fact that there are many good points in the application of power to street car propulsion which are omitted in each of the systems now before the public. An eclectic system, chosen and adapted from these and railroad practice, would, if practicable under present commercial and legal conditions, probably cover the ground better. A modern system, for instance, which is confessedly imperfect in many points, excels all others I have seen in one particular—that of attachment to axles. This is on the differential principle, and admits not only of several changes in relative speed and "purchase" between motor and axles, but of entire disconnection on down grades to rest the motor. A friction gear was used, so as to give the desired changes by the movement of the operating lever.

The coming street car will probably comprise flexible gear with long wheel base, independent wheels on (preferably) tubular axles and power track brakes. Its motors will be differen-

tially connected. It will not only excel old practice in ease of riding and comfort, but will double the present life of wheels, with great economy in track, truck and motor repairs, as well as in power, lubricant, etc. This and more, at schedule speeds of ten to twenty miles per hour and without sacrifice of safety, is now in sight. The public taste and demand will become more exacting as the horse car recedes from view, and managers will more and more appreciate the fact that only the best is cheap when bidding for the good-will and patronage of intelligent people.

DISCUSSION.

Mr. W. S. Rogers.—I feel as though I would like to make an apology on behalf of builders of street cars, although I am not in any way connected with them, or even acquainted with them. The writer states that they "put everything on the car body," and in the present confusion existing among electric companies, I do not wonder at it. The maker of axles does not know just what to make for them. When he gets an order, it is general. The maker of the car body does not know what to do. It is not like building the standard trunk-line cars, where the car is built complete and shipped to the road. So I think the street car builders themselves should be very leniently dealt with in a case of this kind, until the electric companies find out, at last, what they want and what they intend to do. At present it looks to me as though the main thing the electric companies are doing is to eat each other up, and to prevent each other from getting any good improvement and using it. I think, after they have got through with that, that we will have good street car construction all the way through.

CCCLXXI.

GRAPHICAL ANALYSIS OF RECIPROCATING MOTIONS.

BY OBERLIN SMITH, BRIDGETON, N. J.

(Member of the Society.)

THE object of this paper is to call the attention of engineers to a graphical method of representing all reciprocating motions in machines, which have such relations to each other that they must perform certain portions of their journeys in given times. It may also be used to represent rotary motions; especially is this desirable if there be intermittent or variable velocities. I have elsewhere (see "Machinery Designing" in "Mechanics" for April 15, 1886,) described this system in much fuller detail than I will take the time for here. It is a method which may be, and probably is, in use by many others beside myself; but I infer that it is not very generally known, from the fact that I have never seen a description of it in print nor known of its use by any of my friends. It is of the simplest possible description, being nothing more than the plotting of charts, such as are used to show pressures and temperatures in steam engineering, or the relative values of grain and tallow, of dividends and expenses, or of the heights of mountains, and populations of cities, during various successive periods of time, in commercial and geographical engineering.

This system I have used for several years in my own practice, and would as soon think of getting along without it as I would without a clock or an almanac, or, as regards the machines treated of, as I would without the scale drawing showing the shapes and sizes of their various members. It consists simply in drawing straight or curved lines, or a combination of both, which represent the successive positions of any given points in a moving member of a machine. These "time-lines" have a general course from left to right, starting at a zero point representing a point of time which is the beginning of one cycle of the machine in question. This is preferably the "stopped" position in machines which stop at definite intervals, like power presses, etc., but may

be at any convenient point in continuously running machines. The whole space on the chart to the right of a vertical zero line represents *time*, and is divided by faint vertical lines into uniform spaces, representing, of course, equal periods of time. These might be minutes, seconds, or any other known periods, but, practically, it is more convenient to let them represent a given number of degrees of revolution of the main shaft of the machine, or of an imaginary shaft which would revolve once in one cycle of time. The vertical lines are therefore designated by degree numbers running from 0 to 360, their distance apart usually being either 5 or 10 degrees.

All vertical distances from a horizontal base line, produced horizontally from the point where the time-line starts, represent actual distances of *motion* of the "timing-point," as I will term it, whose motions we are analyzing as representing the motion of the machine member in question. These are, of course, the ordinates of the time line, and are sometimes above and sometimes below the base line, as the case may be. They may be drawn to any convenient vertical scale, but it is preferable to make them of a scale of $\frac{1}{1}$, that is, to let them represent the actual scale of the distance moved. This distance is usually counted *in the path* of the timing-point, although sometimes, where such path is a curved one, it is more convenient to give the time in an imaginary straight path represented by the chord of its arc, or in an "altitude" line not following this chord. The timing-point is an assumed point which can obviously be located anywhere in a member that slides in a straight line, but which in oscillating or rotating members must be at some known and convenient distance from the axis of motion.

The horizontal scale of these charts may, of course, be of any proportion desired. In my own practice it usually varies from 6″ to 12″ to represent 360 degrees, 9″ being the most common distance, in which case each $\frac{1}{4}$″ counts for 10 degrees. I usually make them on the same cross-ruled paper which is used in my office for all scale drawings. This is graduated in faint red lines 1″ apart each way, and still fainter lines every $\frac{1}{8}$″. Usually no special ruling is necessary; *e.g.*, if 9″ is used as the horizontal scale, each $\frac{1}{8}$″ of course represents 5 degrees, and the vertical lines running all the way down the chart enable the times of various points in the time-lines to be readily compared. The horizontal red lines serve to show vertical distances, and one of the heavier ones is selected

for the base line of each time-line, the respective ones used being placed at convenient distances apart, in whole inches, to allow for the length of ordinates necessary, so that the time-lines will not intersect each other.

In general, it is well to use a scale which makes the time-lines approximate the average circumference of the majority of the cams used in a machine. The angle then made with a vertical line by the steepest portion of a time-line shows how steep its cam will have to be in certain places, where it has to do the worst "up-hill" work, that is at what angle its periphery will be with a radial line at that point. This, as is well known, should not usually be much less than 45 degrees—as a roller or other surface pushed by a cam will not smoothly mount a steeper incline without undue pressure upon journals, fulcrums, etc. In laying out cams to drive rollers, the path of the roller axis should be considered as the theoretical cam periphery, rather than the actual surface of the cam itself, which may in some cases have steeper slopes than above mentioned.

The name of the member whose motion is to be represented is usually written on the left-hand margin of the chart, and various memoranda are inserted at different points along the time-line to show the functions of the different parts of the motion shown by the respective portions of the line which have different directions.

Any horizontal part of a time-line of course represents a "dwell" or absolute stoppage of the timing point in question; *per contra*, a vertical portion of a time-line would represent infinite speed, and therefore, of course, never occurs. A diagonal portion of the line obviously represents uniform velocity.

Appended below are three diagrams, which are reduced facsimiles of three working charts picked up at random from a number which I had on hand, and which had accomplished their life-work with perfect success in enabling certain newly invented automatic machines to start off successfully as soon as erected without all the chipping, filing, patching, and revolving of cams upon their shafts, which is so frequently practiced in work of this character.

Fig. 31 belongs to a machine for compressing medicinal tablets, where only three principal motions were required. The time-line *a* represents a cam motion. In *b* is shown the characteristic line of a crank motion. In *c* and c^1 is shown a cam motion again, which in this case was made variable in amplitude by the inter-

vention of a lever with a variable radius, arranged by a pin adjustable in a tee-slot, etc., to be from nothing upward. The line *c* represents one extreme where the motion was reduced to *nil*, and *c*' shows it when at its maximum, there being any number of other lengths of stroke between.

Fig. 32 belongs to a tin-scrap-nail machine. The time-lines *d*, *e*, *f*, and *g* depend upon cams, while *h* has a crank motion. The line *i* belongs to a member driven by "*h*" through the intervention of a spring which yields when "*i*" comes against a stationary "stop," thus giving the dwell shown by the horizontal line near the

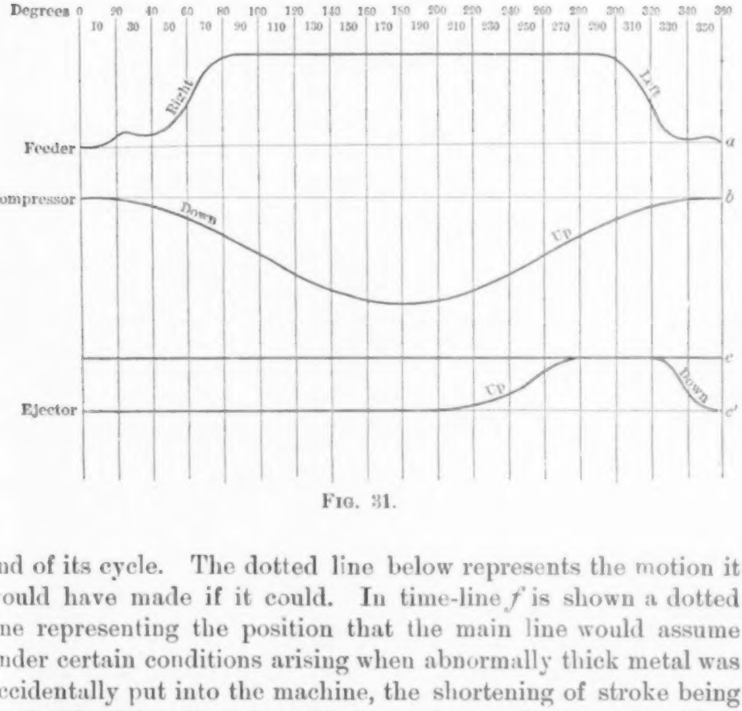

FIG. 31.

end of its cycle. The dotted line below represents the motion it would have made if it could. In time-line *f* is shown a dotted line representing the position that the main line would assume under certain conditions arising when abnormally thick metal was accidentally put into the machine, the shortening of stroke being due to elasticity allowed certain levers, to guard against breaking. In time-line *d* the dotted line is a "wear-line," assumed as representing the probable time of the machine after a given number of years' wear upon the cams, rollers, etc.

Fig. 33 is the chart of a special machine for automatically making electric-wire-cleats. The time-lines *j*, *k*, *l* are similar, because the members concerned are all attached together, while the member for *m* is elastically driven against a stop, as previously men-

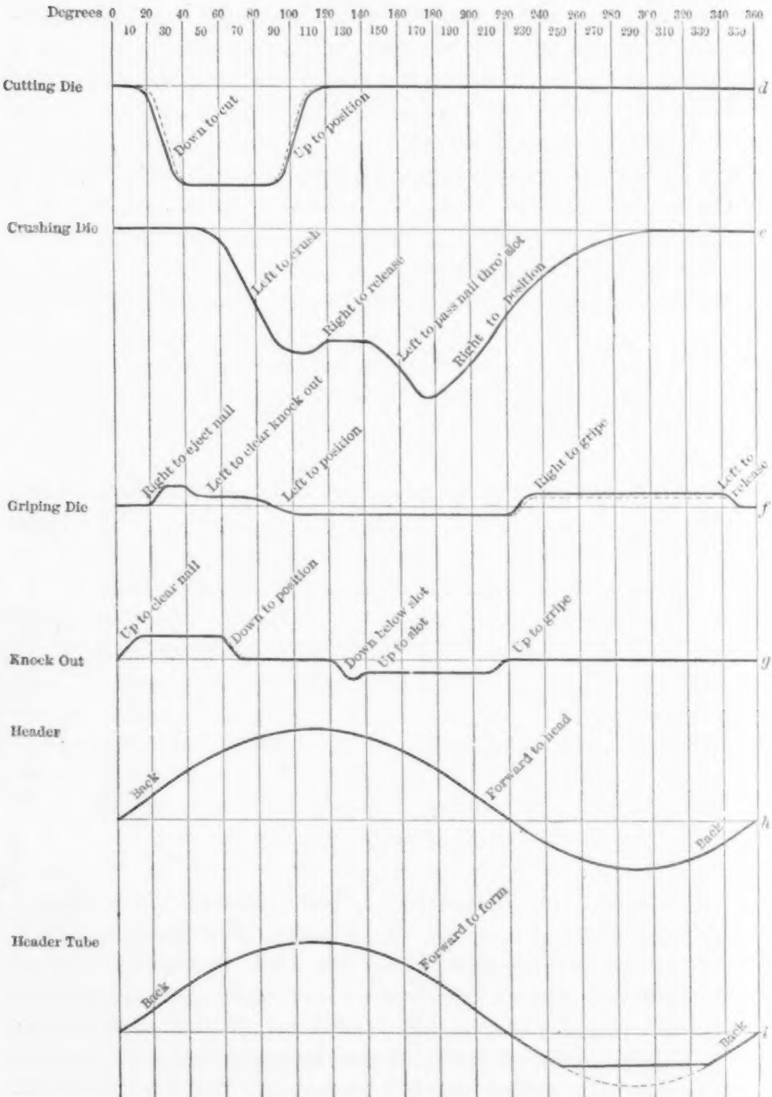

FIG. 32.

tioned regarding i, Fig 32. At n is shown the cam motion of a member which would be timed as in the dotted line (like j) if it were not gripped and held by a device attached to a member moving as at o, etc. The lines o and g show cam motions, while

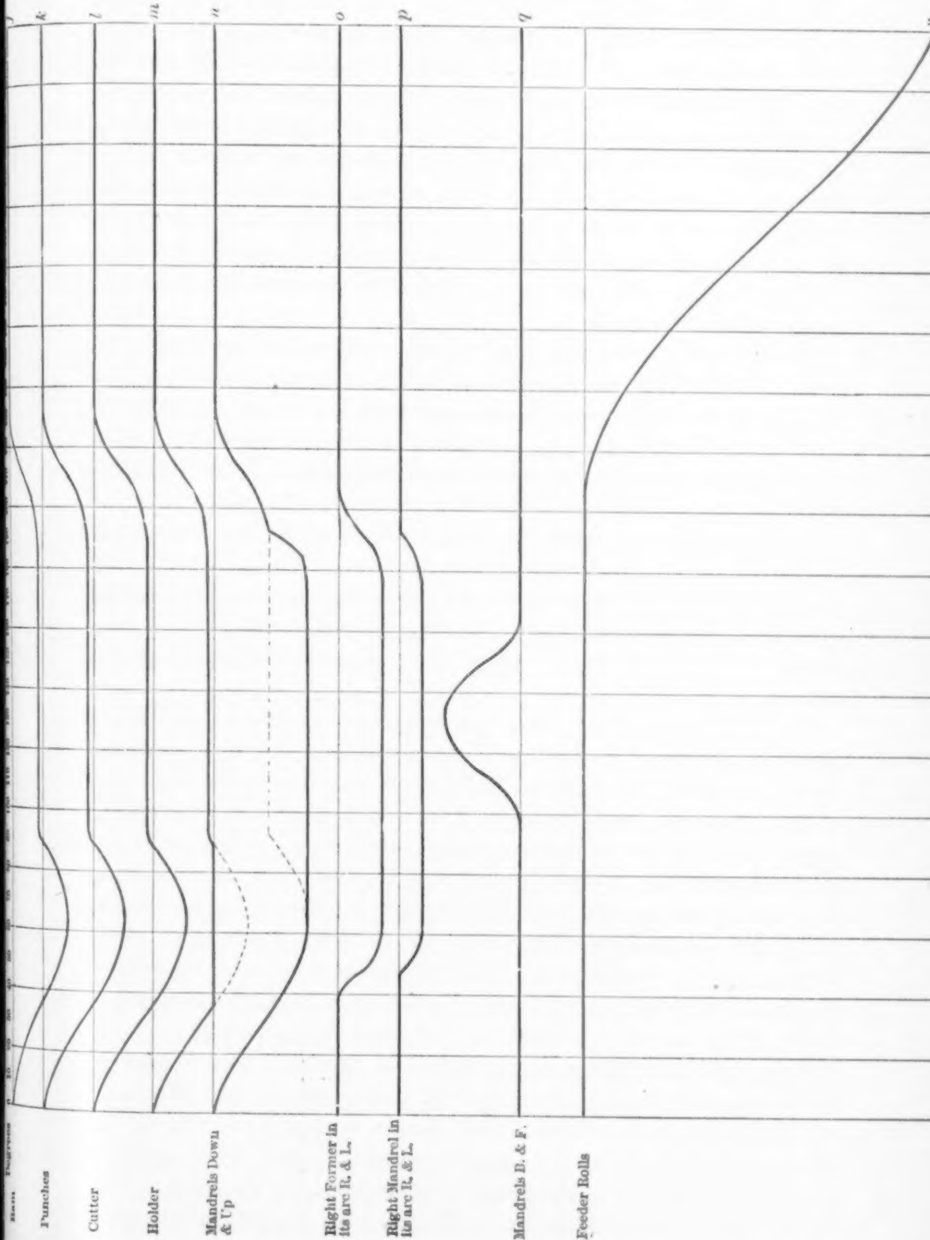
Ram Degrees
Punches
Cutter
Holder
Mandrels Down & Up
Right Former in its arc R. & L.
Right Mandrel in its arc R. & L.
Mandrels B. & F.
Feeder Rolls
k
l
m
n
o
p
q
r

FIG. 33.

p is, like *o*, but modified by a "stop." At *r* is shown a line which, unlike all the rest, does not return to the base line. This represents the motion given to a strip of material while passing through ratchet-driven feed-rolls; also, of course, the periphery motion of the rolls themselves. In this case such a strip moved to the right a given distance and was then cut off, without returning to the left. At the next cycle of the machine this motion was repeated, and so on *ad infinitum.* Such lines would obviously occur in the charts of all machines thus intermittently fed in a continuous direction; as the rod in a rivet machine, the thread in a sewing machine, etc.

The charts shown are not entirely systematic, as they were made by different draughtsmen at different times, and are given only as actual specimens of practical work. I intend in future to systematize them properly, practising more uniformity in regard to the memoranda of directions and functions. It is intended also that the "wear-lines" should be inserted wherever such wear as will occur in a reasonable length of time, say ten years, will injuriously affect the action of the machine. Thus, in analyzing and comparing the motions of the various members, the original time-lines and their successors, the wear-lines, would both be considered in making each motion have a proper relation to the others.

The merits of this system no one who has tried it will for a moment dispute. It permits a cam to be laid out with absolute accuracy, basing its various radii upon the respective ordinates of the time-line pertaining to it, and also shows the necessary proportions for a wedge, lever, toggle, etc., which may be a factor in producing the motion required. It also aids in obtaining smoothness of motion with consequent quiet running and durability, by the facility with which too sudden starts and stops become plainly visible to the eye. Sharp angles in the time-lines, of course, represent jerkiness of motion, with noise and rapid wear, while smoothly rounded curves show the opposite.

This brief description is presented in the hope that it may be the means of bringing more generally into use among Mechanical Engineers this obvious and simple system, one withal the principles of which are already practised in analyzing cylinder pressures, as well as in showing the state of the pork market, and the growth of the population of China. No copyright has been applied for.

DISCUSSION.

Mr. Chas. E. Emery.—Our author has a very happy way, now and then, of stating in a paper the results of his shop practice. I have always felt grateful to him for his paper showing the principles involved in drawing, from plates, vessels of different shapes. The method referred to herein is also described in a very interesting manner, but I must submit that the method, though well adapted, perhaps, for a particular shop practice, does not show the proper way to plot cams for regular use The cams operate on a system of polar co-ordinates; therefore, why not lay them out as such, and not on a rectangular co-ordinate system. The paper mentions that an angle of 45 degrees is about the maximum proper for a roller running on a cam, but, evidently, if a certain amount of radial movement be attained through a certain number of degrees of arc with an angle of 45 degrees, where a cam is of large radius, the same movement through the same arc will require a much steeper angle where the cam is of smaller radius. Frequently, also, the desired movement cannot be obtained in a given arc with size of cam ntended, not only because the angles are too steep, but because a re-entrant curve comes out of less radius than that of the roller. The natural way of laying down a cam is on the drawing of the cam itself, and if other cams are to keep time with it, they can be preliminarily laid down over it, and then transferred to other sheets. An accurate rectilinear motion derived from a crank is represented simply by a circle, as in the Zeuner system of showing valve motions, and the distortions due to short connecting rods, toggles, etc., can be made to modify this circle, as well as the particular curve shown on the rectangular system.

Mr. Geo. M. Bond.—I should like to say that the practice of the company with which I am connected, in cutting cams, is to have the cutter travel practically in the same path as does the roll which is to work with the cam, and in this way to get the relation of the working surfaces as nearly the same as possible in the making as in action.

These "paradigms of motion" have been successfully applied in the design and execution of a model type-setting machine by the Paige Company, in which both the timing and fitting have proved equally successful by the application of this method,

when the machine was assembled. No scraping or fitting was required throughout the entire series, in which a great number and variety of cam motions and timing is involved.

Mr. Jesse M. Smith.—I happen to be working on a machine, just at present, which involves this very question. I find that I want to make a cam on which a roller shall roll properly without any jar, and I want to give it a certain size, so that the friction on the pin about which the roller revolves shall be as small as possible, and, to make it all smooth working, I also want the centre of this roller to follow a certain line. When I come to give the centre of the roller a certain diameter, which is the diameter of the roller itself, I find that in certain parts of the cam the roller is not supported at all. The track on which the roller should travel is entirely cut away, and it is necessary, as Mr. Emery has said, to lay those things out by polar ordinates, rather than by rectangular co-ordinates, so as to find out just exactly that point, and it may be necessary to change the entire form of the cam, so as to get a path on which the roller shall travel.

Mr. F. M. Leavitt.—It seems to me that we are mixing up two things that have nothing to do with each other. I have had to lay out a good many cam-motions myself, and have found it advisable first to determine the motions which it is necessary to give to the various parts of the machine without regard to the shape of cams or other details of mechanism. I have always used this method of Mr. Smith's, and think it is the best plan, because you can better study the relation of parts, detect interferences, and all that. After determining these points, and getting the parts of your machine to move as you wish, you must then go to work and lay out the necessary cams, or other devices, to produce these motions, which is quite a different affair from the first. It is simply a question of detail of construction, and has nothing to do with the timing of the machine.

Mr. H. W. Spangler.—It seems there is only one thing which the author means, and that is simply what the last speaker says. He has no idea of giving you any means of laying out the cams, but simply giving you the proper motion at the proper time. The scheme itself, if I remember rightly, is, to a certain extent, outlined in a book that, I suppose, now is considered obsolete—"Willis' Elements of Machine Designing." A fair idea of the principle can be got from that, but, of course, it is in a crude

shape. The underlying idea, however, is very clearly set forth in it.

Mr. H. H. Suplee.—I had in mind the same writer to which Mr. Spangler refers, and I think that Professor Willis there stated that this idea was first struck by Mr. Babbage in designing his very complicated calculating machine, and that he found it necessary to plot in some way the various motions in one diagram, in order to have them agree.

Mr. Oberlin Smith.—It is very evident that the first three gentlemen who spoke have not studied this paper, or, if they have, they do not comprehend it very well. Mr. Leavitt, however, seems to understand perfectly what we are driving at. The paper does not attempt to describe the process of laying out cams. I have no doubt we should be very glad to hear, some time, as to Mr. Emery's methods, which probably are very much like those of the rest of us. We, of course (in my own draughting room), use the circular method in *laying out* cams, and find out whether a certain assumed proportion with which we start will answer. Very often, as another gentleman has observed, the cam resolves itself into nothing, and at certain points we have to make it larger in average diameter all over; or else we change the slope at the troublesome point, thus getting a different time of motion, in order to make it a practical cam, because some of it has tried to disappear. Of course we find such difficulties when we come to lay out the cams themselves, but, as Mr. Leavitt says, when we begin we must represent the proposed motions somehow. I think this is often done inside of some one's head, and some exceptional men can thus do it. It is, however, something like playing four or five games of chess at once, blindfold. Now, this chart is exactly to the motions of the machine in question what a chess-board is to a game of chess. Without it, the majority of designers must go a good deal by the rule of "cut and try."

The system described is a practical one, and has long been in use by me with the help of ordinary draughtsmen. We do not, moreover, any of us know how we could get along without it. On our preliminary chart we see that one thing does not interfere with another by running down the vertical lines. We thus tell at a glance where a certain member has come at a certain time, and whether the other ones will dodge it. If there is more time than is necessary at one point, we can shorten it a certain

amount of time, and lengthen it out at another place. And so we go over the thing until we get a set of motions which seem, on the whole, to be best. It is usually a very short job, too. Then the next thing we do is to lay out the cams and other driving devices according to this chart. It is easily done, because all we have to do is to take our vertical ordinates from the chart, and transfer them to radial ordinates on the proposed design, or, rather, the "roller-path" of the same, as Mr. Emery would appear to suggest.

Referring to what Mr. Bond said, regarding cutting cams, this is exactly my own practice. I have also used a grinding wheel in the same way as his cutter, to finish off cams which had to be very accurate, keeping the grinding wheel of exactly the same diameter as the future roller.

Mr. Spangler evidently sees the thing just as did Mr. Leavitt, with a correct conception of what it means. I was not aware of anything of this kind having been published by Willis or Babbage. Of course, I shall be interested in looking it up, but I understand from one of the speakers that they did not work it out very completely. I have not given my system as new (although original with me), but simply as a good, practical method of working, which I know is not in very general use.

CCLXXII.

METHODS OF REDUCING THE FIRE LOSS.

BY C. J. H. WOODBURY, BOSTON, MASS.
(Member of the Society.)

THE liability to injury by fire is a hazard inherent to all buildings, and a constant menace which imposes upon the owner a persistent outlay, which endures as long as the building stands.

As every method of construction, the various mechanical processes, and the stock in each stage of manufacture each bear some relation to the fire hazard as a supporter or possible originator of combustion, the engineer whose duties pertain to these matters must necessarily consider not only the direct application of those engineering problems required in the design and installation of fire apparatus, but also the question of the fire hazard in the important phase of prevention.

The fire loss is a most oppressive tax, much of which can be abated by the application of well established means of prevention. In a practical sense certain fires are to be considered as unpreventable, being caused by exposure to fires in other burning buildings, and therefore entirely beyond the control of the injured party. Other fires proceed from causes so rare that they are not anticipated, and in any event it might not be feasible to prevent their occurrence.

There are, however, very few fires whose destructive results might not have been prevented by the exercise of precautions entirely feasible in their nature.

These several topics will be considered in reference to the reduction of the fire loss on isolated manufacturing property, because the exercise of every possible precaution may not avail anything if the property is liable to be imperilled by fires originating in adjacent buildings.

SUPERVISION.

Care is the most important element in the prevention and extinguishment of fires. Only a small proportion of fires is the

result of a direct act of an individual, but they proceed from natural causes, and might have been prevented by the exercise of due precautions.

An experience-table of mill fires for over a generation shows that the largest fires have not reached their destructive results through the absence of what could be considered adequate apparatus, but by the lack of care or suitable management of such apparatus.

The prevention of fires must in greater measure proceed from the efficiency of the supervision exercised. This must include inspection of the buildings, heed to probable causes of fire, and attention to the fire apparatus.

In a manufactory there is a wide distinction to be made between to-day's dirt and yesterday's dirt, meaning by the first the rubbish necessarily incident to manufacturing, and by the second its neglect. Nearly every kind of bye-product is liable to spontaneous ignition, and should be removed to a place of safety before night. This precaution does not apply merely to oily waste, but to every kind of waste material, even including iron turnings, whose oxidation when wet is a frequent cause of fire. It is not safe to make a discrimination in regard to oily waste, because there are numerous fires starting in what is considered, and truly appears to be, clean waste. The best waste boxes are made of galvanized iron, like small ash cans, except that they are provided with short legs, so as to stand about four inches above the floor. The waste cans, after being emptied at night, can be overturned in the middle of the floor; any infraction of this rule to be reported by the watchman.

The best administration of an establishment does not appear to be that where the manager tries to do everything, so much as it is where he does not do anything that some one can do just as well for him—perhaps better. While the manager of a large establishment should be conversant of all details pertaining to that position, it is sometimes better for him to supervise certain details by knowing the deviation from the rules and policy laid down by him, rather than by an endless reiteration of matters which are strictly in accordance with a routine, and therefore requiring no special action on his part.

BLEACHERY AND DYE WORKS.

INSPECTORS' WEEKLY REPORT.

The undersigned report the result of their examination of the different departments as follows :

DEPARTMENTS.	Casks and Pails.	Rags and Dirt-Boxes.	Hydrants.	Hose and Fittings.	Closets and Benches.	Fire-Doors.	Dirt around Steam Pipes.	Gas and Water-Pipes.	Pumps tried or turned.	Steam-Pipes.	Elevator Machinery.	Elevator Hatches.	Safety Lamps for Lighting Gas.	Unclean Machinery.	Lanterns.	General Order.
Colored Finishing Room	o	o	o	o	o	o	o	o	o	o	o	o	o	o	—	o
White Finishing Room.	o	o	o	o	o	o	o	o	o	o	o	o	×	o	—	o
Calendar Room No. 1..	o	o	o	o	o	×	o	o	o	o	—	—	o	o	—	o
" " " 2..	o	o	—	—	o	—	o	o	—	o	—	—	—	o	—	
Can " " 1..	o	o	—	—	o	o	o	o	—	o	—	o	×	o	—	
" " " 2..	o	o	—	—	o	×	o	o	—	o	—	—	—	o	—	o
" " " 3..	o	o	—	—	o	o	o	o	—	×	—	×	—	o	—	o
Winding Room........	—	o	—	—	o	—	o	o	—	o	—	—	—	o	—	o
Grey Room............	o	o	o	×	o	o	o	o	o	o	—	—	×	o	—	o
Box Shop..............	o	—	o	o	o	o	o	o	o	o	—	—	o	o	—	o
Kier Room.............	—	—	—	—	o	—	—	—	—	o	—	—	o	—	—	—
Bleach Room..........	—	—	—	—	o	—	—	o	—	o	—	—	o	×	—	—
Starch Room..........	—	—	o	—	o	—	o	o	—	o	×	—	×	o	—	o
Dye House No. 1.......	—	—	o	—	o	—	o	o	o	o	—	—	o	o	—	o
" " 2.......	—	—	—	—	o	—	o	o	—	o	—	—	—	o	—	o
Logwood Shed.........	—	—	—	—	—	—	o	o	—	o	—	—	—	—	—	o
Wheel-Houses.........	—	—	—	—	o	—	o	o	o	o	—	—	—	o	—	o
Machine Shop..........	—	—	—	—	o	—	o	o	—	o	—	—	o	o	o	o
Carpenter Shop........	o	—	—	—	o	—	o	o	—	o	—	—	o	o	—	o
Dry Sheds..............	—	—	—	—	—	—	—	—	—	—	—	o	—	—	—	—
Boiler Rooms..........	—	—	—	—	o	—	o	o	o	o	—	—	—	—	—	o
Store House............	—	—	o	—	—	o	—	—	—	—	—	—	—	—	—	

REMARKS.—We would again call attention to boxes used in Colored Finishing Room as being a source for collecting inflammable material. Would suggest that a metallic box of some kind be used in Grey Room to hold oil-cans, coal-tar, and lead mixtures ; also, that a *pan* be used for holding oil-cans in Box Shop. We would also suggest for consideration the closing of fire-doors in store-house automatically.

JOHN WINN,
C. J. MULLANEY, } *Inspectors.*

In order to have a full knowledge of the details of a large establishment, some method of inspection is necessary, and while this will differ in details at every establishment, yet there is much which is common to all. Let the inspection of the whole property be made on Saturday afternoon by two men—such as foremen or overseers of rooms—who may be appointed to serve four weeks, their assignment terminating on alternate fortnights. The report should be made on a sheet of paper divided into squares, each of the horizontal lines being devoted to one of the rooms of the establishment, and the vertical lines dividing it into columns for marking the condition of the order of the room, machinery, and fire apparatus, in the several details as shown by the report inserted

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on the previous page. Approval being marked by a circle, fault by a cross, and the absence of the feature in the column by a dash, the completeness of the report is assured by a mark in every square, and the result of the whole work is clearly shown on the chart, taken from an actual report, in a manner similar to a roll call.

Although fires originate more frequently during the day, yet the absence of the employees to render service causes the greater destruction to be done at night. It is of the utmost importance that the work of the watchman should be carefully arranged.

The watchman should be a strong, efficient man, and not, as is frequently the case, employed for the position because he is good for nothing else, and can therefore be hired cheaply. He should understand how to start the fire pumps, as well as to manage any part of the fire apparatus. As property should be watched during the day Sunday, as well as at night, it is under the care of watchmen about five-eighths of the time, and the measure of this responsibility should be clearly understood.

The overseers should remain in their rooms after the help have left at night, and should be joined by the watchman, and not go away until both are agreed that the room is in proper condition in regard to care of waste, windows, steam, and water, and a record has been made on the watch clock by each of them.

The route over the works should be carefully laid out so that the watchman should go through and not merely across every room, and at least once an hour go out of doors in order that the sense of smell should be more acute to the faint odor which precedes spontaneous combustion. In many instances a dog is a very valuable ally for a watchman.

The best method of furnishing a watchman with light in buildings is by means of a gas jet near each end of the rooms; but, as this is rarely practicable, lanterns must be used, and especial care must be given to the construction of lanterns. Safety requires that the lanterns should be securely guarded; that the handle and sustaining parts of the lantern be connected together by rivets or by locking the metal together without relying on soldered joints; and thirdly, that the lamp should be put in from above and never from the bottom.

The patrol should be recorded on a watchman's clock, not merely to show that he was not unfaithful, but also to prove that he was faithful. There are a great number of varieties of watchmen's record clocks. In all of them the record is made by marks

or perforations upon a sheet of paper, generally circular, which is revolved once in twelve hours like an hour hand, by means of a clock work. There are radial lines which indicate the time, and they are generally provided with concentric circles upon the paper, which designate the number of each of the stations visited by the watchman. The mechanical clocks, sometimes in portable form, and sometimes fixed in each room, are being supplanted by the electric clocks, of which there are a great number of types,—all of them good, but at the present time preference seems to be given to those in which the electricity is generated in each of the stations by means of a magneto-machine, which is operated by the act of the watchman in making the record. This arrangement obviates the necessity of batteries, and as there is no electricity in the system except at the instant of its use, the opportunities of fraudulent record appear to be impossible. Especially in districts liable to disorder and lawlessness, it is desirable to have a district messenger signal box in the works, visited once an hour, with the understanding that if the call is not made within fifteen minutes of the appointed time, it will be assumed that there is trouble and help sent at once.

In other methods, electricity serves a useful purpose in devices for the protection against fire. The telephone and the city fire alarm system are considered indispensable wherever possible. The auxiliary fire alarm in connection with the city circuits helps to save the seconds which are so precious at time of fire. A large amount of ingenuity has been expended upon various forms of automatic fire alarms, in which the heat will expand or fuse metals, or volatilize a liquid, and this in turn actuate electrical contacts and in that manner produce a fire alarm. The general experience with these automatic fire alarms has not been as satisfactory as could be wished, on account of their liability to get out of order, and in an inoperative condition, or to give false alarms with undesirable frequency.

There are numerous minor electrical devices, such as an alarm and record whenever a fire-proof door is opened, or an alarm whenever a main bearing is becoming warm. All these electrical devices are apt to require the supervision of a skilled electrician to maintain an assurance of their operative condition.

Whenever there is a necessity of night or Sunday repair work requiring the use of any artificial light, more than one man

should be employed, and the same precautions as those concerning watchmen's lanterns apply here.

The lack of proper lubrication, particularly on main bearings, is a fruitful cause of fire, which calls for careful attention that such journals be kept in line, and freely lubricated with good oil.

The storage of oil should be outside, in a building either below the surface or banked around; and in one instance the oil house is placed near to the sand bins of a foundry. The most stringent measures are necessary to prevent oil being drawn when artificial light is used.

It is necessary not merely to see that steam pipes are free from contact with combustible material, but also that they are not covered with a non-conducting material which will ignite at the temperature to which they are subjected. Some of the materials used for covering steam pipes cannot be set on fire until heated, and then they become highly combustible.

The number of fires originating from the contact of combustible or oily material with steam heating pipes has been very materially diminished by a return to the earlier custom of placing the pipes overhead, hanging them in coils of about four pipes placed about three feet from the walls and two feet below the ceiling. Although this method of heating is very efficient in keeping the temperature at the desired point and the room more uniformly heated with a smaller expenditure of steam than is required by the ordinary method of heating a room by means of steam pipes placed next to the walls and near the floor, yet the later method of warming a building by blowing air which has been heated by passing through a case containing the coil of steam pipes is now considered the most desirable and efficient means of heating a building, and in this connection is commendable on account of its freedom from fire hazard.

The class of fires occurring in the process of lighting up a mill are not readily prevented, as the work is necessarily done by the help employed, and accidents will occur even when jacket lamps are used with a netting around the light. It is the relief from the hazard of lighting the burners which has rendered electric lighting the safest method of artificial illumination.

Certain classes of material, such as lime and powdered zinc, are liable to produce fire when wet, and should receive due care on this account.

The hazard from storage of raw material or finished goods is far less than that of the manufacturing processes; and careful attention should be given to the desirability of keeping the manufacturing buildings as free as possible from any material except that which is in the process of manufacture.

In the settlement of losses from destructive fires, the interests of the assured have frequently suffered through the lack of a schedule of machinery, giving an appraisal of the contents. This is especially true in regard to patterns, which may pertain to machinery for which there is a constant demand, or which may belong to machinery long since superseded, or which may be a portion of some machine made for an inventor or manufacturer, charged for in the bill at the time and representing neither value nor investment. One of the largest and best managed shops in the country uses the following method of dividing up the book containing the appraisal of patterns, and a similar method for all stock:

Progressive No.	Shop No.	Quantity.	Description.	Weight.	Rate.	First Valuation.	Last Valuation.	Discount.	Present Valuation.

Another method is a portion of a very complete system of accounting in the establishment of a manufacturing stationer.* Forms similar to the enclosed are used with certain modifications for stock in process, or articles which will be for sale, as the articles which are in the following register are not for sale.

* J. C. Blair, Huntingdon, Pa.

JANUARY, 1889.

A

Rough Register Inventory. ______________________ 188__

ENTER ON THIS BLANK ALL ARTICLES
NOT INTENDED FOR SALE
EXCEPT PIPING.

Building.......... ... . Floor..................... Called by.. ....

Room.. .... Section........... ...... Entered by ....

In taking Inventory use only the Columns Marked with a Star (*) and under no circumstances fill in amounts, etc., in columns not so marked.

Entry No.	* Register No.	* Quantity.	* Description.	* Size.	* Imp. No.	Purchased from.	Date of Purchase.	* List Price.				Fr't Charges.		TOTAL NET.		ACTUAL COST. Registered or belonging to that Class, (Cost, etc.).		ACTUAL COST. Attached or belonging to that Class, (Cost, etc.).		Appraised Value, 1889. Registered.		Appraised Value, 1889. Attached.	
								Dollars.	Cts.	Discount.	Cost Net.	Dollars.	Cts.	Dollars.	Cts.	Dollars.	Cts.	Dollars.	Cts.	Dollars.	Cts.	Dollars.	Cts.
866																							
867																							
869																							

COST TOTALS: TOTAL FOOTINGS, - Am't carried from preceding pages, Am't forward to next page, - -

APPRAISED TOTALS: TOTAL FOOTINGS, Am't carried from preceding pages, - - - Am't forward to next page, - - - - - - -

A simpler method, and probably just as complete in many instances, is the one given below, and is used for appraising a larger amount of property than either of the other forms.

No. of Item.	Floor.	When Built.		Description of Item.						Total.
					1st Value.	Dep.	Net Value.	Setting Up.	Amount.	

Another duty of the manager of the works, which is too often overlooked under pressure of immediately urgent affairs connected with production, is that of supervision of the fire apparatus, both in regard to its care and also practice in its use ; and the neglect of such supervision constitutes a deficiency which is responsible for many unexpected losses.

Fire apparatus should be kept in service as well as in order. It is no exception to the rule that practice is essential to obtain efficient results. During the warm season of the year a portion of the help should be formed into a fire organization, arranged by assigning a man to each part of the fire system where there is liability of needing any help, and this organzation should practice from time to time, at least twice a month, in order to familiarize themselves with the use of the apparatus, and also to reveal any defects in its installation.

The details of such organizations differ with the arrangements and administration of every mill ; but the general policy of definitely assigning persons to the positions for which they are best adapted and where it is presumed they could be most useful, and to practice them in such work, is a rule which is common to all. In establishments like steel works, where the operation is continuous, it is necessary to have an organization made up from each of the gangs of men. By way of a distinctive feature, the members of one fire organization are provided with belts with a special clasp, and when any person wearing one of these belts leaves work, he is to give up his belt to his alternate before leaving the yard. In this manner by retaining the wearers of the belts always on the premises, it is arranged that there shall always be a fully manned fire organization present during the working hours of the week. It is not generally desirable that overseers or room foremen should he assigned positions in such organizations, as it is clearly their duty, in case of fire, to remain if possible in the rooms under their supervision.

The practical results of such fire organizations, where fire has occurred, have been very marked, and systematic and skilful work has been the rule in place of the needless confusion and liability to breakage of fire apparatus which almost inevitably occurs where there is a lack of such organization.

It is also essential that pumps should not be kept for fire purposes only, but that some arrangement should be made so that a pump, even if its service is not required for other purposes, should be put into operation at least once a week.

The arrangement of pipes should be as simple as possible, and especial care taken that the gates in the mains should be marked with an arrow showing the direction of opening, and that the drip valve for the purpose of removing the water from the pipes of the system should be in a prominent location, instead of a concealed and hardly accessible place, as is too frequently the case. The valves to automatic sprinklers should be sealed open by placing a strap around the pipe and one of the spokes of the wheel of the valve and securing the ends together by a rivet or belt cement.

A great deal of fire apparatus is destroyed by freezing water during the winter months, and therefore a special inspection of all such apparatus should be made late in the autumn, when the water should be drained from all portions of the system where there is liability of freezing, and all hydrants and valves should be well oiled, preferably with mineral oil. The hazard should a hydrant or other portion of the apparatus be broken by frost does not lie so much in the probability that disadvantage may result from the disuse of one element of the plant, as in the liability that such a breakage may interfere with the whole system and render it inoperative.

Construction.

In its design, a mill for any standard line of manufacture is not a building whose arrangement and proportions are fixed upon at the whim of the owner, but they must conform to certain conditions of dimensions, stability, light, and application of power to satisfy the requirements essential for producing the desired results at the lowest cost,—circumstances and conditions to which capital has been mercilessly starved by close competition and the encroachments due to growing socialistic tendencies. As methods of manufacture change, so mill construction

must in like manner be altered to conform to the new conditions.

Whatever changes may be made in methods of construction, the question of the fire hazard continues to be represented by an annual charge which differs from the other fixed charges of interest in that its ratio is a variable one and is based upon the estimated annual chance of fire. The more efficient the manner in which such a building can be arranged to resist fire, by so much will the hazard represented by the insurance premium become less.

Although a great deal depends upon construction, this hazard can never be reduced to zero, as that would indicate that the mill was absolutely fire-proof. A fire-proof mill, both in name and reality, must not only be built of incombustible material, but must also be capable of resisting any fire of its contents without destructive effects upon the structure. Such a building would be a commercial impossibility, both in regard to prohibitive costs and also the hindrances to manufacturing involved by a method of construction which would bear comparison with the casemates of fortifications, and is therefore unsuited to the conditions essential for the advantageous operation of machinery. It is apparent that the name of fire-proof mills never deceived anybody, for the owners keep them insured, and the insurance companies charge a rate in accordance with their estimate of the hazard.*

The destructive consequences attending fire in such buildings, whose iron and masonry construction is called fire-proof, shows that some other form of construction is necessary to obtain the desired result of minimizing the annual cost of the maintenance of the invested capital, as represented by insurance, depreciation, interest, and taxation. There is little incentive for entering into unusual expenses in the construction of a manufacturing building for the purpose of increasing its resistance to fire, unless the additional interest on such increase in the investment is to be met by a corresponding reduction in the annual cost of the fire hazard. In addition to these questions involving the annual

* William A. Green, when Chief Engineer of the Boston Fire Department, in answer to an inquiry from an official at Berlin, asking for a list and description of the fire-proof buildings at Boston, wrote that the Beacon Hill Reservoir was the only fire-proof building in Boston, although they did not at times feel quite sure of it.

maintenance of the plant, the increase in the expense of the building above a certain point may prove poor management by locking up capital for too long a time, and may tend to prevent the improvements in arrangement and construction which are necessary for the most advantageous manufacturing.

The method of mill building known as slow-burning construction combines the advantages of low initial cost and great resistance to destruction by fire, the final result being that the manufacturing process is housed at the minimum annual cost. Such slow-burning methods of construction also furnish the most advantageous arrangements for manufacture as to stability, light, and application of power; the whole being the result of the best fitness of means to ends in the effort to reduce the cost of production to its lowest terms.

The principal features of slow-burning construction may be illustrated by bits of experience drawn from many sources. The fundamental principle of such construction is to mass the material in such a way that there shall not be any concealed spaces about the structure, and that the number of projections of timber which are more easily ignited than the flat surfaces shall be reduced as far as possible: that iron portions of the structure shall not be exposed to the heat of any fire in the contents of the building, and furthermore that the isolation of the various portions shall be as complete as is feasible—both as respects one building to another and the various rooms and stories of the same building to each other.

It is a rare occurrence that one has an opportunity to arrange an establishment on an extended scale from the very beginning. In such instances it is easy to organize the whole so that the dangerous processes shall be relegated to special buildings, that the storage shall be separated from manufacture, that the progress of stock in process of manufacture shall be continuous, that due preparation shall be made for future extensions—and particularly in regard to the arrangement of buildings to grade—and that full arrangements for the application of power may be made so as to take advantage of the various possible economies from the latest state of the art. Under such circumstances the arrangement of all pipes and fire apparatus can be made to the most efficient purpose.

Far more engineering skill is required for the reconstruction of old establishments, bringing their methods as far as possible

Author requests that the Note herewith may be pasted opposite the cut, Fig. 34 on page 283 of Volume XI, Transactions of the American Society of Mechanical Engineers.

NOTE.—The figures on the right hand wall indicate the thickness at the panels in line with the windows and not the thickness of the pilasters.

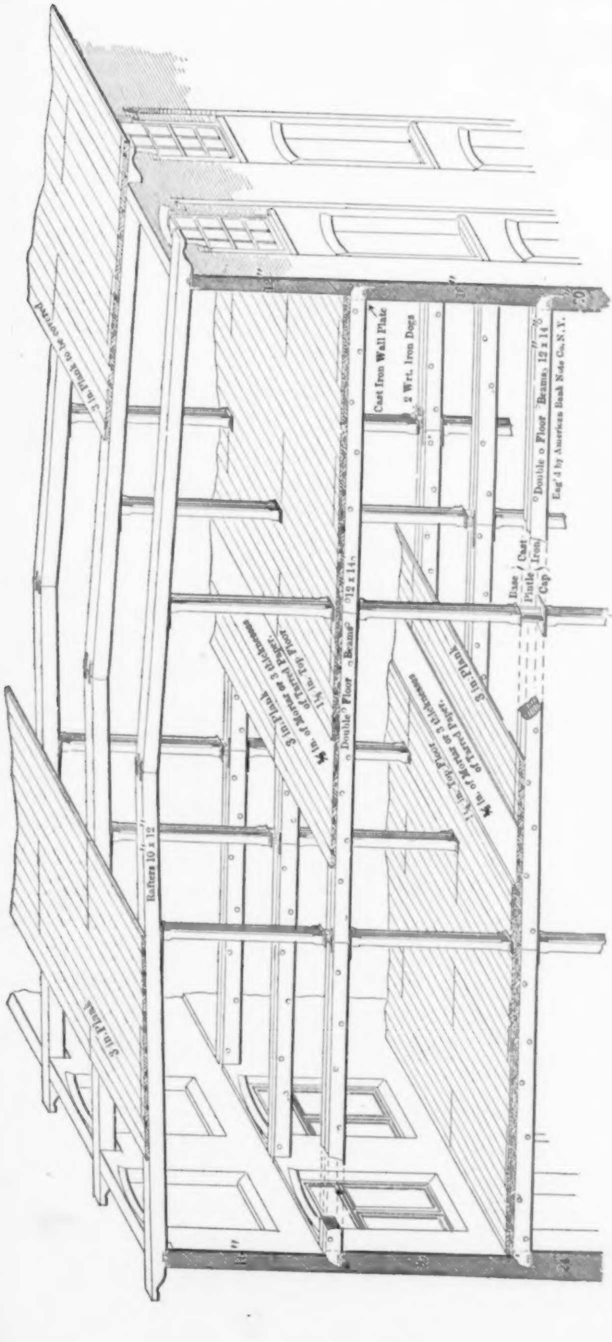


FIG. 34.

SLOW BURNING CONSTRUCTION.

to the present state of the art; and in many instances this work has been done with but little interference with the production.

Some of the more salient features of modern mill construction pertaining to questions of resistance to fire can be briefly related, without entering upon the general subject of mill construction. The most important feature is that of the mill floors, which should be laid on beams, generally of southern pine, 12 x 14 inches, or two inches larger when required by unusual loads or longer span than 22 feet (Fig. 34). These beams are placed from 8 to 10 feet apart between centres, and it is preferable that they should be made up of two pieces bolted side by side, with small blocks interposed to provide an air space of about an inch, as this will diminish the tendency to decay or to twist as the result of long continued seasoning. At the anchorage of the walls the beams should rest upon cast iron plates secured into the walls and provided with a rib on the top one and a-half inches in height projecting into a wide groove across the bottom of the beam and brought to a firm bearing by a pair of wedges driven into a groove each side of the iron tongue. It is important that an air space should be left in the wall each side and at the end of the beam to prevent dry rot, and that the bricks in the wall for about four rows immediately above the beam should be laid in dry sand. A preferable modification of this is to use a cast iron box made to receive the end of the beam and separated from contact on the top, end, and sides of the beam by ribs projecting on the inside of the box. As in the former case these iron supports are securely built in the wall. This arrangment is much more stable in anchoring the beams to the wall, and if by reason of fire the beams break they will slide out from their anchorage without injury to the wall as soon as the groove on the bottom of the beam clears the rib in the supporting plate.

At the columns, beams rest on cast iron caps which should present a supporting area at least three times that of the cross section of the wood columns used, as the resistance of timber to longitudinal crushing is three times that of its resistance across the grain. The support from one column to the next should be made by cast iron pintles, preferably those whose section is in the form of a Greek cross, as that presents advantages in the way of securely joining them to the timber beams. At the top of the pintle a cast iron plate should support the base of the

column above. A flat plate with a projection in the middle securing the column by insertion into the core of the column is preferable to the cup form which is so frequently used, as the latter tends to retain moisture to the deterioration of the column by dry rot.

Timber columns are preferred to those of iron, unless the load is greater than can be sustained by timber,—the limitation of a safe load on oak or southern pine of straight grain and free from knots being about six hundred pounds to the square inch. Such columns, when subjected to destructive tests, give way by direct crushing, and for this reason the tendency of late years has been to make square instead of round columns. These offer no greater obstruction to the floor than the round columns and give about one-fourth extra resistance. Wood columns should have a $1\frac{3}{4}$ inch hole bored along the axis, and two $\frac{1}{2}$ inch holes through the column near to each end, to diminish checking. Whenever the amount of load renders it necessary to use iron columns, they should be protected by wire lath and plaster, or by some of the special tiles made for the purpose.

The floor planks for this type of floor are generally made of spruce plank from three to four inches in thickness grooved on both edges and joined together by hard wood splines. These floor planks should be two bays in length, breaking joints at least every four feet.

Above this the top floor, of $1\frac{1}{4}$ or $1\frac{1}{2}$ inch hard wood, is laid; and in some instances the resistance of the floor to fire is greatly increased by laying a coat of plaster on the floor plank before the top flooring is put on. But the general method of increasing the resistance of the floor to fire is to cover the floor and beams on the under side with plaster laid on wire lathing. Whenever anything of this kind is done care should be taken that the covering to the timber should not be hermetically sealed, as that course increases the tendency to dry rot, particularly if the timber is not very thoroughly seasoned at the time of its application. There have been a number of failures of the strength of such construction on account of the disregard of this well-known principle: that it is not desirable to seal up the outside of unseasoned timber. This not only applies to the method cited above, but also to paints and varnishes which are laid on such work before it is thoroughly seasoned.

In mill floors of very large area care should be taken that the

transverse shrinking of the floor should not pull the walls; and on large low structures which are far enough across to be affected by a very slight longitudinal shrinkage of the beams, other methods of anchorage than the one referred to should be adopted.

The mill roof is similar to the floor in many of its characteristics, the timbers being somewhat lighter. If the roof is higher in the middle the beams extend through the wall, and their ends properly cut serve as brackets for the roof planks, which are continued to the ends of the beams, and thus make a finish without calling for the construction of hollow cornices with gutters.

There will be no trouble from condensation on such a roof in cold weather if the plank and roof boarding are three inches in thickness, unless some wet process be carried on which produces a large amount of vapor, like paper manufacturing; in this case it is advisable to make the roof four inches in thickness, also placing roofing felt between the plank and the roof boarding.

If it is desired to add finish by placing sheathing against the underside of the roof plank, such finish should be blind nailed, as the nails are good conductors of heat, and reaching to the colder portion of the roof, the moisture in the air will condense on the nail heads if exposed. It was formerly assumed that a double, sloping, hollow roof was necessary over a paper machine, but the above type of roof has been successfully employed for such purposes.

A coal tar concrete walk about the building will protect the foundations and also take care of water dripping from the roof. Under some circumstances the slope of the roof is reversed, being depressed at the middle and highest at the edges, the storm water being removed through gutters extending down through the middle of the building. This enables the walls to be extended above the roof, forming a parapet which is a great defence against fire from exposure outside, and is a method of construction which is being generally followed in cities.

There is but little to be offered in this connection on the subject of walls; but the method of building pilastered walls is becoming more frequent, as it affords an opportunity to dispose the brickwork more rigidly than in plain brick walls, and also to introduce features of architectural symmetry. With buildings one or two stories in height, walls are made of plank in a manner

similar to the mill floor, producing at low cost a building which is strong, warm and light, without also containing the combustible features of an ordinary frame building.

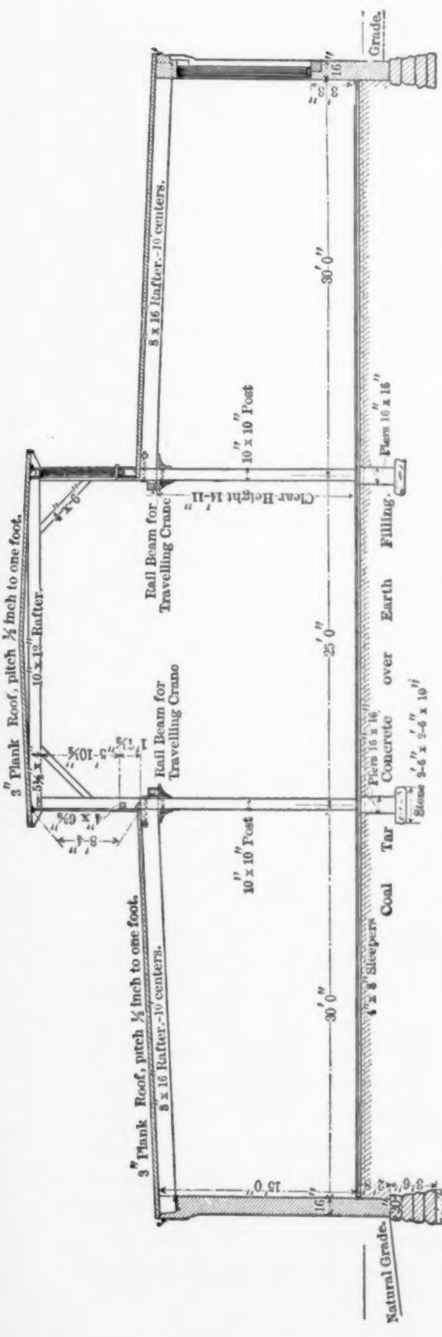
Such a mill floor and columns, while possessing in a very high degree features which offer resistance to fire, and are weakened only to a slight extent as they are slowly burned away under exposure to a very severe fire, also possess the merit of great economy, both as regards the low price of construction, and in that the floor is thinner in comparison with joisted floors of equal strength, saving in this respect for every floor in a building about ten inches in height of wall, stairs, belting, steam pipes, and all vertical connections reaching from floor to floor—a saving which amounts to considerable in the total cost of building.

It is important, however, that floors should not be pierced with unprotected openings for stairways, elevators, or main belts. There is no record of such a mill floor, without belt or other holes in it, being burned through by a fire starting in the room below.

The principles of mechanics to be followed in the design of a mill floor are not those of the ultimate resistance to breaking, but of the amount of distortion which it will be permissible to impose upon the machinery by reason of the yielding of the floor. The usual method of fixing upon the limit of such deflections as a constant ratio of the span is clearly wrong in principle, the form of the flexure being a curve, and, therefore, not a constant ratio of the span. The proper allowance, however, of the measure of the amount of distortion allowed to the floor is that of the mean radius of curvature—a safe limit being that the deflection in inches should not exceed .0012 multiplied by the square of the span in feet; the desired result is to fix upon conditions of stiffness of the floor plank and the floor beams, which will cause it to deflect to an equal radius of curvature in both directions by the same load per square foot.

Floors near the earth in basements should be well ventilated, if there is a space between the floor and the surface of the earth; but, unless that space is continuous and two or three feet in height, it is better to lay the floor directly upon the earth, taking care to prevent dampness by putting in a layer of stones or cinders. Above this the sills for the floor are laid in a trench with coal tar concrete, which is also laid between the sills, on a level with the top of them, and the floor boards are laid over

FIG. 35.



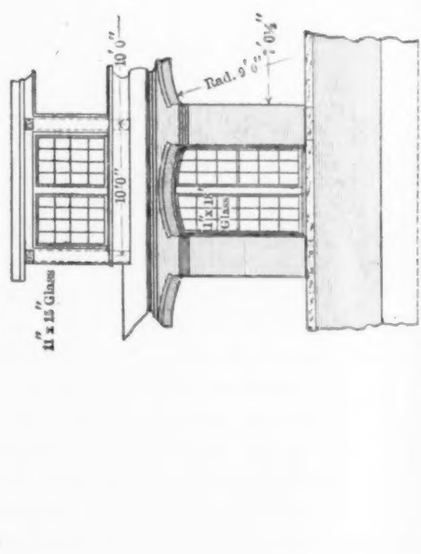


FIG. 36.

ONE STORY MACHINE SHOP.

this. When well-seasoned chestnut is used for this purpose, it will last a great many years without dry rot. Cement concrete will cause timber to decay rapidly. The sills are not necessary except to furnish material to secure lag screws holding down quick-running machinery; and where the two courses of floor plank are sufficient for such purposes, the sills may be omitted and the plank laid directly upon the concrete.

Windows for a mill should be placed as high as possible, as the illumination from the upper part of a room is much more uniformly diffused. The window sills should be sloping, in order that articles should not be left on them in a disorderly manner. The architectural effect is usually heightened by placing sashes as near to the line of the inside of the wall as possible, making the disclosure or opening around the sash on the outside as large as possible. Ventilation can generally be arranged more satisfactorily by means of stationary sashes, the upper part of which swings like a transom over a door. In places where the work will be disturbed by currents of air, a desirable form of window ventilator is made by deflecting a portion of the middle of the sash—about eighteen inches in height and of same width—inward into the room, covering the top of the bracket-like projection thus formed with a cover which can be opened at will. The air from the outside is directed upward toward the ceiling, thus giving a uniform circulation in the room without causing rapid currents. In case the windows admit too much light, this can be readily modified by painting the inside of the windows with a mixture of turpentine with zinc white. The amount of obstruction desired can be readily fixed upon after a few trials.

In one-story mills, and in the upper stories of wide mills, it is necessary to introduce light through the roof. While the expensive English ridge and furrow roof gives a very soft and blended light, it is hardly adapted to the northern part of the United States, on account of the difficulties with snow, and monitor roofs or hipped skylights give the best satisfaction. If the latter are used, it is desirable in many instances that the lower part of the space under the skylight be separated from the room below by means of glazed sashes, placed flush with the ceiling. The monitor roof of a one-story machine shop affords an opportunity for a travelling crane with a minimum of building space (Figs. 35 and 36).

Stairways should not be placed in direct communication be-

tween different stories of the mill, but in masonry towers, with substantial doors at each story. Although almost generally a better architectural effect can be produced by building such stairway towers on the outside of the mill, yet in many instances principles of economy can best be served by building the tower inside of the mill at a corner. Stairways should be made in straight runs, with square turns rather than spirally. Foothold on the stairway may be increased either by laying upper treads containing a number of grooves, similar to those frequently used on the steps and platforms of street cars, or, what is better, by

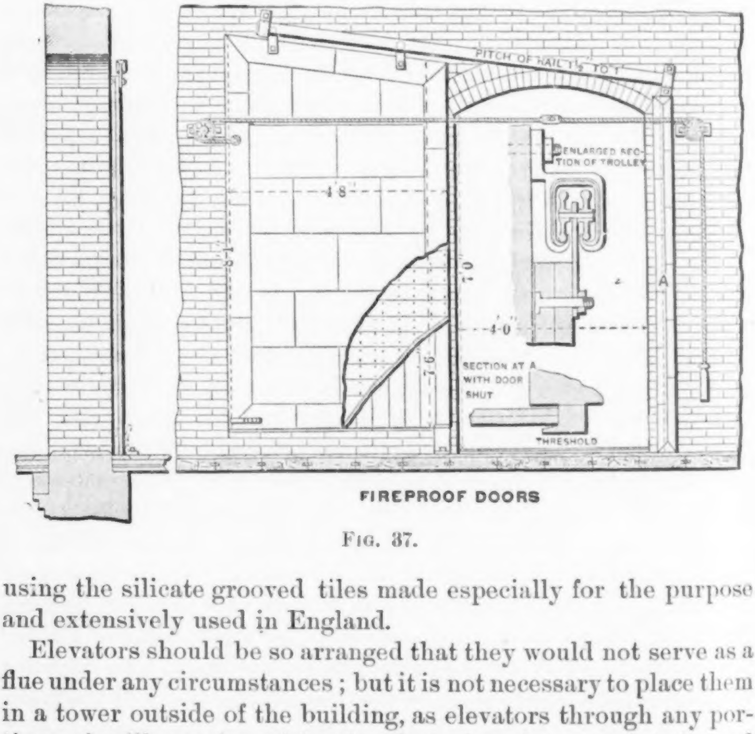


FIG. 37.

using the silicate grooved tiles made especially for the purpose and extensively used in England.

Elevators should be so arranged that they would not serve as a flue under any circumstances ; but it is not necessary to place them in a tower outside of the building, as elevators through any portions of mills can be safely placed at almost any point desired for sake of convenience, and the openings kept closed by means of automatic hatches.

The division of mills into various portions by means of fire-walls is frequently not so efficient as assumed, by reason of the lack of fire-doors to fulfill satisfactorily the purpose of resisting fire. The best form of fire-door is that made of two thicknesses

of matched boards, placed at right angles to each other and nailed together, being covered on the outside by tin, securely locked together and held to the door by numerous hanging strips (Fig. 37). The door should be secured to the hangers by means of bolts, and not screws, and the rail upon which it runs should be strongly bolted to the wall. When closed, such door should fit into a jamb and be securely held in this manner against the wall. Such doors are frequently hung upon an inclined track, and, by some application of highly fusible solder at the catch, are so arranged that they will be closed by the heat of a fire, if not closed by hand. This same principle may be applied to fire-proof shutters hung over various openings which necessarily exist in walls which are desired to form a fire-proof separation between various portions of a building. A fire-wall must extend through the roof, cutting off continuity of woodwork, especially at the roof cornices. The top of such a fire-wall can be covered with tiling, which is made in special form for such purposes, better than by a stone coping.

In this treatment of the arrangement of buildings to resist fire, consideration has not been given to the cost of land, which is of itself an important factor in determining what arrangement will be the most expedient for an establishment. Where land is expensive, or where there are limitations in the space suitable for building, it is frequently necessary to build mills and shops higher than would be preferred under other conditions; but where circumstances will permit it, the one-story mill has been very successful, not merely in immunity from fire and very low cost per square foot of floor, but also in the advantages of manufacturing, particularly in regard to cost of supervision and movement of the stock in process of manufacture. These are questions which must be determined, not merely in regard to the various processes of manufacture, but the individual needs of each concern; the position of the fire risk in the matter being that the hazard of a building increases very rapidly with its height and to some extent with its area. The extension of one-story buildings over too large an area will not be commended, and certainly, as regards the question of fire, it has a tendency to place too large a property in direct exposure to a very wide hazard.

The illustration (Fig. 38) is taken from a photograph of a one-story cordage factory designed by Stephen Greene, Mem. Am. Soc. Mech. Engineers. This mill was erected in place of a three-

story building like the one shown in the background of the picture, which had been destroyed by fire, but the cost of manufacture was so much less in the new mill than in the old mill that the latter could not compete with it; and, having outlived its usefulness, was taken down as a cumberer of the ground in order to afford an opportunity for the extension of the one-story mill.

The train in the foreground is drawn by a locomotive operated by compressed air, its service displacing the work of twenty-two horses and sixteen men.

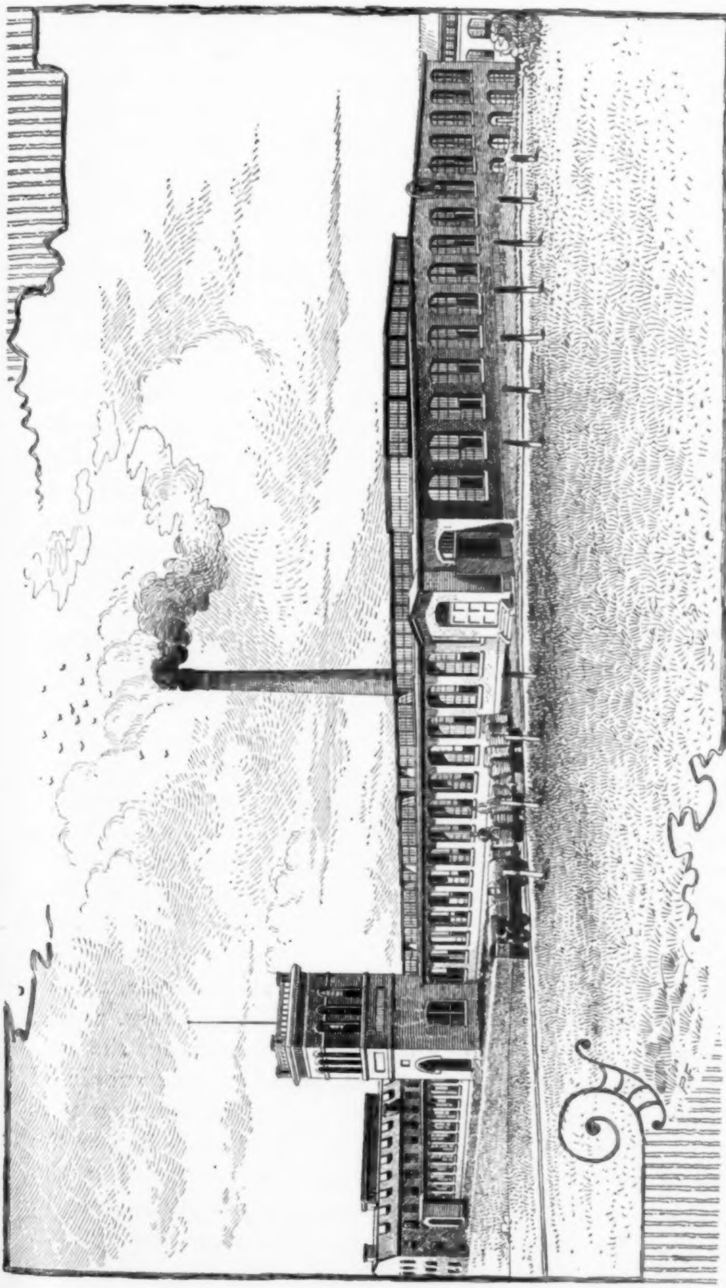
Some textile mills have been built in the form of the block letter U, this form having been decided upon as giving the conditions of lowest resultant cost. One wing, two stories in height, contains weaving; the other wing, three stories in height, contains carding and spinning; while the engine is placed in the connecting building. The pickers and the boilers are in outside buildings, so placed that they will not interfere with future extensions of the building into the form of the block letter H.

In this connection there is but little reference to be made to storehouses, except to mention the necessity of the closest supervision of their contents. The preferable arrangement of stairways and elevators for such structures of more than one story in height is to place them in towers upon the outside of the building, not directly connected to the rooms, but reached by means of open galleries which lead from the door of the storehouse to a corresponding door in the tower at the same level. The height of each story should be low enough to prevent overloading. The best conditions of safety require that not over one row of stock in bales on end should be placed on a floor. It is very important that the floors of storehouses should be made as tight as possible and provided with scuppers with swinging covers on the outside to discharge any excess of water thrown on them in case of fire.

Machinery.

The actual hazards from the tools and machinery used in manufacturing processes are so many that there is no opportunity for more than passing reference to some of those of quite frequent occurrence.

In all machinery hot journals are a prolific source of fires, and protection is to be secured by free lubrication with oil suited



From a photograph.

FIG. 38.

ONE-STORY MILL.

to the especial service required of it. In many instances it would be a good investment to place long lines of heavy shafting in pillow blocks on masonry piers, rather than in hangers or in harness frames. This particularly applies to the methods of transmitting power to the beating engines in paper mills.

Taking the dangers from machinery in the order of their causes, those from prime movers require the first consideration. The dangers from fire in boiler rooms are few, the risk being well understood and is looked out for accordingly. If other perfectly well known, although not as apparent, possibilities of fire were as carefully guarded, there would be a great diminution in the number of fires.

The fires from the spontaneous ignition of bituminous coal are very difficult to conquer. The explosion of gas generated by the hot coal frequently causes serious damage in addition to that from the direct combustion of the coal. Such fires are prone to occur when least expected and to cause unwarrantably large damage. The cause of the spontaneous ignition of coal is undoubtedly the heat produced by the chemical changes occurring in the inorganic sulphur compounds abounding in the coal; these changes being favored by time, dampness and pressure. While it may be assumed that within certain limits the percentage of such inorganic sulphur compounds would be a measure of the hazard of spontaneous ignition likely to arise from any quantity of a given coal, yet there are difficulties in the way of an application of any such assumption. The sulphur which causes the damage is not that of the average of the whole mass, but the maximum in any lump of coal exposed to favorable conditions; and secondly the analyses generally give the total percentage of sulphur without separating the organic from the inorganic compounds.

Returning to the more practical features of the matter, bituminous coal should not be stored under valuable buildings, nor in contact with timber. When a lot of coal is used up, the coal shed should be thoroughly swept out before a new supply of coal is added. If the depth of coal exceeds eight feet, iron rods should be pushed to the bottom and examined every day, as the warmth of any heating within the pile is conducted to the protruding end of the rod, and thus an opportunity is given to shovel over the coal before it actually reaches the ignition stage.

There are no precautions to be suggested for anthracite coal storage, as the hazard is almost nominal.

The general use of crude petroleum for fuel is too new to fur-

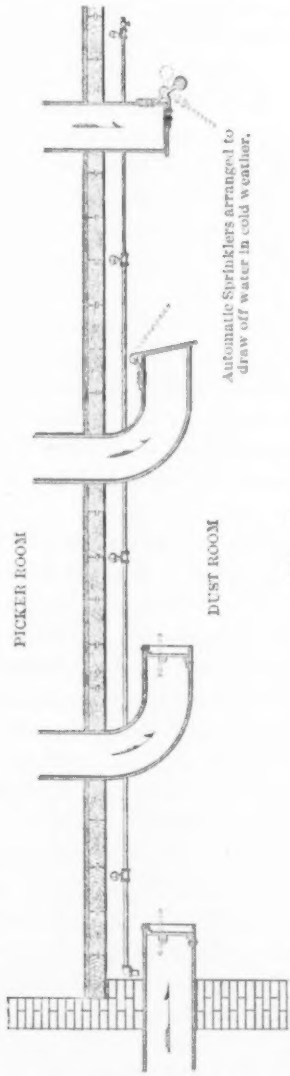


Fig. 39.
AUTOMATIC DAMPERS FOR DUST FLUES.

nish a history, but the record thus far has contained accounts of many accidents, principally in connection with faulty methods of storage. The petroleum should be securely stored in venti-

lated, covered, iron tanks, placed below the point of consumption, and where the oil cannot endanger any property in event of breakage. At the point of consumption, the oil is attended with the hazards of other gaseous fuel, particularly at the instant of lighting, and in like manner it is necessary to apply the flame before turning the oil spray on to the furnace.

The crude petroleum used for fuel has a flashing point of 16 to 35 degrees Fah., giving off inflammable vapors at about the freezing point of water. The reduced oil has been "treated" in order to remove the readily volatile material, and its flashing point raised to 125 degrees and upward, sometimes reaching 300 degrees Fah. It is much safer than crude petroleum, but very difficult to pump in cold weather.

In the steam engine room, hazards are not due to the machine, but to neglected oily waste and similar reprehensible matters.

Water wheels are subject to dangers in connection with the wearing away of the wood step, which will occur at odd intervals and without any previous notice; the alteration in the height of the turbine shaft frequently causing the bevel gears to strike fire, thus igniting the grease.

The distribution of power has been attended with many and severe fires, generally resulting in what is known as a total loss. The greater part of this destruction may be avoided by building a belt tower in the mill, opening toward the engine room and covered by a glass roof with wire netting beneath the glass. The driving is in this tower, and the power is transmitted by shafting passing through holes in the wall of the tower.

All machinery which throws a waste product in a dust room, as cotton pickers, buffing machines, and napping machines, frequently ignite such fluff, and the fire returns to the room where it started through the flue or trunk of some other machine not in operation, or the flames burst from several of the machines when they are stopped at noon or night. Such mishaps are materially reduced by placing light swing dampers at the ends of the flue where they are kept open by the current of air from the blowers of the machines in operation, and at other times are shut (Fig. 39).

The floor under machines liable to scatter oil should be protected by sheet metal, particularly in the case of woolen cards, mule heads, and in some instances electric motors.

The limitations of a single paper will not allow further allu-

sions to the dangers from particular machines; but the principle of keeping them in the best operating condition is also that of keeping them in the safest condition, and no one is better fitted for the exercise of such care for the prevention of fire than those engaged in the practical attendance of such machines.

Fire Apparatus.

All methods for the prevention of fires fall so short of the ideal of immunity that there is a necessity for fire apparatus. The principle of defence of a manufactory against fire is that of self protection by making the installation and management of the fire apparatus of such a grade as to be able to cope with the progress of any fire which can possibly occur. The merits of fire organizations have already been considered as essential to the service of fire apparatus.

Buckets of water are the most effectual fire apparatus. They should be kept full and distributed in liberal profusion in the various rooms of a mill, being placed on shelves or hung on hooks, as circumstances may require. In order to assist in keeping them for fire purposes only, they should be unlike other pails used about the premises, and in some instances each pail and the wall or column behind its position bears the same number.

It is a mistake to keep fire pails in dry-rooms, as the water in the pails evaporates rapidly, and also in so doing interferes with the drying processes. The pails should be placed in some convenient situation near to the dry-room, where they will not oppose the drying process, and will also be more accessible in case of fire than when hung inside of a dry-room.

In unheated buildings the contents of fire pails can be prevented from freezing in winter by adding chloride of magnesium to the water.

Galvanized iron pails are better than wood pails, and indurated fibre makes a very satisfactory pail, especially in places around bleacheries, chemical pulp, or paper mills, where corrosive fumes rapidly injure metal pails.

There are various expedients to insure the full condition of fire pails, such as various floats, or electrical contrivances, or sealing over the top of the pail some thin sheet of impervious material; but the fact is that there is no fire apparatus so simple and effective as a full pail of water in good hands. All

automatic devices are not above contingencies, and they lead to lowering the standard of personal espionage, which is the controlling principle in the administration of affairs.

Generally there is also need of casks of water to furnish a further supply to the fire pails.

Garden hose attached to a supply of water often constitutes a very useful portion of the fire apparatus. Any cocks in the nozzles should be fixed in an open position by striking a heavy blow on the handle of the plug cock commonly used in such fittings.

Automatic sprinklers have proved to be a most valuable form of fire apparatus, operating with great efficiency at fires where their action was unaided by other fire apparatus, particularly at night. In mill fires the average loss for an experience of twelve years shows that in those fires where automatic sprinklers formed a part of the apparatus operating upon the fire the average loss amounted to only one-nineteenth of the average of all other losses. If the difference between these two averages represents the amount saved by the operation of automatic sprinklers, then the total damage from the number of fires in places in which automatic sprinklers are accredited as forming a portion of the apparatus, has been reduced six and a quarter million dollars by the operation of this valuable device.

Although there have been numerous patents granted to inventors of automatic sprinklers since the early part of the present century, yet their practical use and introduction has been subsequent to the invention of the sealed automatic sprinkler by Henry S. Parmelee, of New Haven, Conn., about twelve years ago. This device being the first, and for many years the only automatic sprinkler manufactured and sold, and actually performing service over accidental fires, to him belongs the distinction of being the pioneer and practically the originator of the vast work done by automatic sprinklers in reducing destruction of property by fire.

Although nearly or quite two hundred thousand Parmelee automatic sprinklers have been installed, their manufacture has been supplanted by other forms; and the total number of automatic sprinklers in position at the present time must be about two million.

In an automatic sprinkler system the sprinkler heads are

attached to tees in pipes against the ceiling; the arrangement being such that there shall be at least a sprinkler to every one hundred feet of floor, some places requiring a still larger number of sprinklers. There should be two sources of water supply, with check valves in the pipes leading into the sprinkler system, giving it the benefit of the greater pressure without the intervention of any personal act. If one of these supplies is furnished by an elevated tank, the minimum head from the bottom of the tank to the highest sprinkler should be not less than twelve feet.

The inability to withstand freezing temperatures is a defect in automatic sprinkler systems which has not been fully remedied by invention. There are many so-called dry pipe systems, in which the water is kept from the system until fire occurs, when the heat which releases the sprinkler is presumed to actuate devices which open the main valves admitting water to the system. Such apparatus is always complicated; these systems have sometimes proved to be inoperative at fires; and have frequently been discovered to be out of order when examined. The attempts at making a solution of low freezing point, which should be non-combustible, and under the conditions of its use should also be non-corrosive, do not appear to have been successful.

Water is sometimes removed from automatic sprinkler systems during cold weather by pumping in air to a pressure sufficient to displace the water. This method demands a great deal of attention; and in case of a fire it requires even longer to discharge the compressed air from the pipes and throw water on the fire than would be the case with the usual dry pipe system.

The only resource for automatic sprinklers in rooms liable to temperatures below the freezing point appears to be to shut the supply valve and slowly draw the water from the pipes late in the autumn and to admit the water in the spring. The valves should be in a place accessible at time of fire, and all persons liable to have any duties in the matter should be made acquainted with the necessity of opening such valves in time of fire.

The discharge of automatic sprinklers, including the resistance of the pipe fitting may be represented by $Q = 0.6\sqrt{p}$, in which Q equals the discharge in cubic feet per minute, and p the pressure in pounds per square inch.

The following standard of sizes for pipes for automatic sprinkler installations is based upon the principle of using the nearest

commercial sizes permitting a uniform frictional loss through the system.

Number of Sprinklers.	Diameter of Pipe.
115	4 inches.
78	3½ "
48	3 "
28	2½ "
18	2 "
10	1½ "
6	1¼ "
3	1 "
1	¾ "

When automatic sprinklers were first introduced there were many apprehensions that leakage, and also excessive water discharged upon small fires, would be sources of damage. In England this opinion found expression in increased insurance rates in buildings where automatic sprinklers were installed.

Many automatic sprinklers have been made in such a manner as to impose unusual stress upon the fusible solder, which is a weak alloy possessing but little resilience, and therefore ill adapted to withstand the forces due to water pressure, water hammer, and what is sometimes greater than either, the initial tension in setting up the sprinkler to make it tight. It is not surprising that such sprinklers break or leak; but among the score or more automatic sprinklers on sale, it is easy to select several varieties, any one of which would impose but little risk of leakage from water pressure.

The logic of figures shows that this liability to damage is merely nominal in the case of well constructed sprinklers. An association of underwriters who have given careful attention to the subject obtained the facts that out of 514,071 automatic sprinklers which had been in actual service on the average for five years, under a water pressure reaching in some instances one hundred and eighty pounds to the square inch, but averaging sixty-nine pounds to the square inch, there had been only fifty-eight instances of sprinklers leaking from water pressure, and three hundred and seventeen instances of leakage from other causes than fire, generally by accidents to the machinery or by carelessness of the employees; the average damage from all these causes being $2.56 per plant per annum.

Although automatic sprinklers have proved to be so reliable and effective, yet, in order to provide for all possible contingen-

cies, their introduction should not displace other forms of fire apparatus, particularly standpipes in the stairway towers with hydrants at each story. The hose at these hydrants should be festooned on a row of pins, or doubled on some of the reels made especially for such purposes. Standpipes are not recommended to be placed in rooms or on fire escapes; and inside hydrants should not be attached to the vertical pipes supplying automatic sprinklers.

One pound of burning wood produces sufficient heat to evaporate six and one-half pounds of water, and, owing to the waste, a much larger proportion of water to fuel is necessary to quench a fire.

Fire pumps are generally too small for the work required of them, five hundred gallons per minute being the minimum capacity recommended. For a five-story mill there should be an allowance of two hundred and fifty gallons per minute for an effective fire stream through a 1⅛-inch nozzle, and for lower buildings the estimate should rarely be less than two hundred gallons for each stream.

Contrary to the general assumption a ring nozzle is not so efficient as a smooth nozzle, the relative amount of discharge of ring and smooth nozzles of the same diameter being as three is to four. For standpipes ⅞-inch nozzles are recommended, but for yard hydrant service the diameter should never be less than 1-inch, and 1¼-inch generally fulfils the conditions of best service.

It is important that the couplings on the hose and hydrants should fit those of the public fire department. The best diameter of hose is 2½-inches; the loss by friction under equal deliveries of water being only one-third in a 2½-inch hose of what it is in a hose 2 inches in diameter.

Fire pumps should be equipped with a relief valve and also a pressure gauge, and placed where they will be accessible under all circumstances, and so connected that they can be started at least once a week.

The best location for fire pumps is a matter differing with the conditions of each mill, but they should be situated as near to the source of supply as practicable, with full size suction pipe, easy of inspection and not containing any avoidable bends.

In a steam mill, it is sometimes preferable to draft the water from a point below where the water of condensation is discharged into the stream, as there is less freezing there; in mills driven

by water-wheels it is a convenience in time of repairs for steam fire pumps to draft water from the wheel pit. Rotary fire pumps should have a short draft, but not placed below the level of the supply.

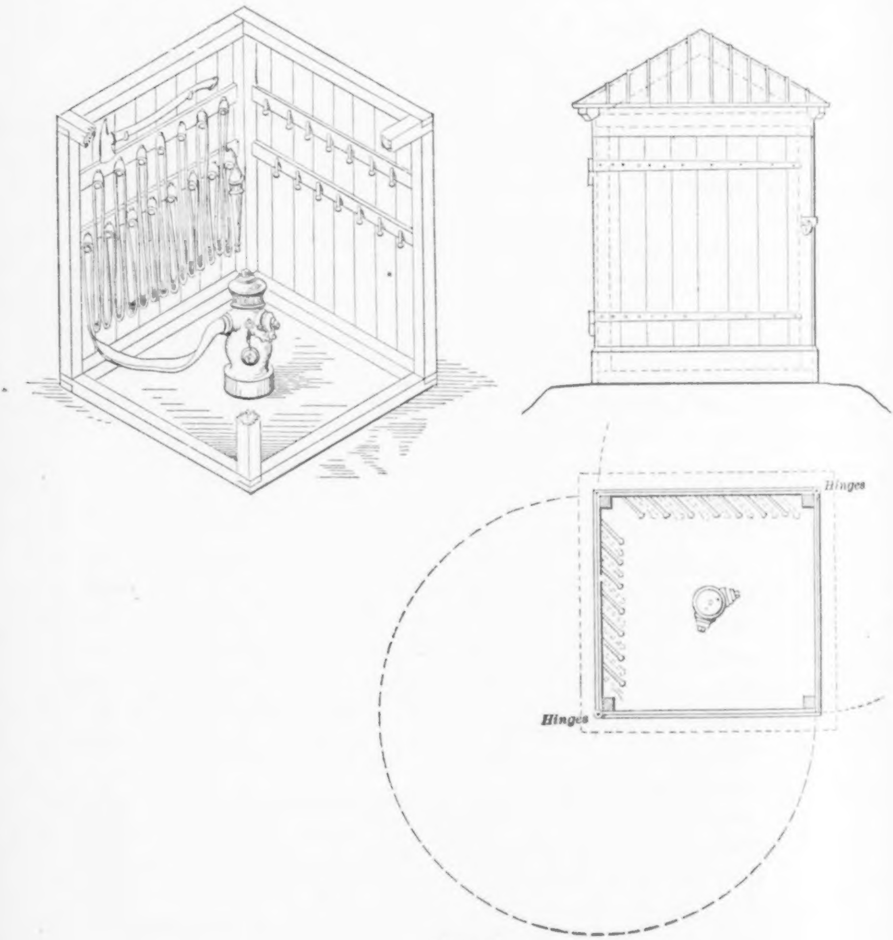


FIG. 40.

HYDRANT HOUSE.

Water mains about a mill yard should be of ample capacity not to cause an excessive loss by friction, their diameter being based upon a limit of velocity of ten feet per second for the maximum delivery.

The yard hydrants should be placed at a distance of fifty feet from buildings, and covered with a house which should also contain hose, nozzles, axes, bars, and spanners. Hydrant houses are made in a great variety of forms, but it is important that the doors should be high enough to avoid ice, or that the house should be placed upon slight mounds. An economical hydrant house may be built six feet square with two adjacent sides hung on hinges, so that the doors can be swung around to the other side and be held by catches. The pins on which hose is hung should be two inches in diameter, and placed diagonally and staggered in two rows. If there is no hose cart, the reserve hose can be placed on shelves.

Stop valves in the mains should be covered by boxes four feet in height, and the direction of opening clearly marked on the hand wheel of the gate.

Results.

These methods of supervision, building, and equipment do not refer to any ideality, but to measures which have been widely carried into effect for the purpose of reducing the fire loss; the result of such action being to diminish the cost of insuring industrial property engaged in such normally hazardous processes as textile manufacture and other industries, down to a yearly cost of less than one-fourth of one per cent. This has been accomplished by the consideration of sources of danger and their abatement, and by a course which has been in line with sound engineering principles, and also practical methods of manufacture; and it has thus been proved that it is cheaper to prevent a fire than to sustain a loss.

There has been no attempt made to credit individuals with their share in these features of mill development. They have been the outgrowth of a continual profiting by experience, adopting some features and modifying others. The concurrent action of the large number of minds engaged on the same problem has led to duplication of methods; but the whole progress has been a matter of slow, steady growth, advancing by hairs' breadths, as the result of persistent efforts to adapt means to ends in the endeavor to reduce the cost of manufacture.

DISCUSSION.

Mr. Frederick Grinnell.—The author has failed to make it quite clear as to the length of time during which was effected the

saving by automatic sprinklers to which he alludes. He speaks of saving six millions of dollars. Of course, to make apparent the value of an apparatus, he ought to couple with that statement the number of properties in which fires have occurred and the time during which this six million dollars' saving was effected.

One statement which is made would convey, I think, rather a wrong impression. The danger or liability of water damage is alluded to in the use of automatic sprinklers in property in general, and it is stated that statistics were obtained to ascertain what this probable loss had been during a given period of years. The statistics which were obtained show a loss of two dollars and some cents per annum for each sprinkler plant or installation. This average loss was obtained by dividing the aggregate loss, which a large number of parties reported as having taken place in their premises, by the whole number of installations of automatic sprinklers. Now that total number of installations included, I think, some four or five different kinds of sprinklers, and of the total number certainly three-fourths were of one single kind of sprinkler which is acknowledged to be specially free from any danger of leakage or of breaking open. Therefore the low average loss obtained by taking into account the large number of installations, where there had been practically little or no loss, would give a wrong impression as to the relative liability of loss with the different sprinklers which were in use, although it is correct as representing the general liability of loss with all apparatus used.

Mr. Scott A. Smith.—In reference to the sketch of fire-proof

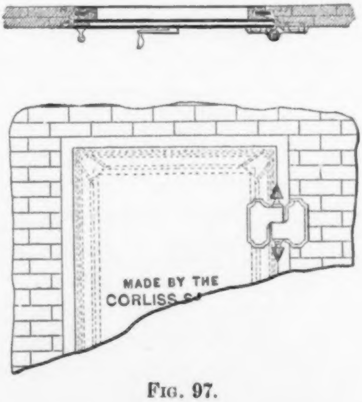


FIG. 97.

doors, I think I may mention something of interest—that the Corliss Safe Door & Vault Company, of Providence, make a double door of two plates of wrought iron, which, when it is shut, has movable flanges, which latter are thrown out into recesses around the four edges of the doors, making a practically air-tight joint through which hot air cannot go (Fig. 97). They have made a large number of these doors for fire-proof vaults for banks and bankers. No doubt it is a practical door for mills and workshops.

Mr. W. S. Rogers.—There is one paragraph in Mr. Woodbury's paper which strikes me with a great deal of force, and it is where I did not expect to see it—in a paper on such a subject as the "Reducing of the Fire Loss." He says: "I think the best administration of an establishment does not appear to be that where the manager tries to do everything himself, so much as it is where he does not do anything that some one else can do just as well for him and perhaps better." That recalls a little incident. I once went into a shop where there was a sign which had been put there by the old superintendent, and it said, "These buckets must not be used for any purpose except fires only," but there was not a bucket in the shop. If he had not tried to see to all that himself, and had left it to some one who could have done it better, the buckets would probably have been there.

Mr. James McBride.—I notice another suggestion in the author's paper regarding the organization of a fire department among the employees of the premises, and that they should be drilled under the supervision of foremen of different departments. I recall one instance where a large manufacturing establishment in the city of Brooklyn had its employees drilled as a fire department, and the proprietor gave exhibition drills there very frequently to show his friends how quickly he could put out a fire. Finally he had a fire, and his volunteer department, instead of flying to their posts, flew out of the building and his whole place was consumed. He then made application to the Fire Department, and adopted every possible means he could, such as telegraphs, signal boxes, etc., for the purpose of getting protection from the department. He thought it was better to depend upon the men who were paid for doing fire work than to depend on his own employees.

Prof. J. E. Sweet.—I think Mr. Woodbury will thank me for

20

calling the attention of the Society to one point set forth in his paper, which is, the importance of having a sufficient supply of water. There was a wagon-shop at Syracuse, supplied with automatic sprinklers, and one day a fire took place. The automatic sprinklers had reduced it so that no blaze was visible. The Fire Department came on, coupled their hose to the hydrants, threw all the water their mains would supply on the outside of the brick walls, while the fire raged within. In other words, the Fire Department took the water away from the automatic sprinklers, and the building burned to the ground.

Mr. Henry R. Towne.—Professor Sweet's reminiscence prompts me to relate another, as illustrating the possibility of having *too much* water in case of fire. In my own establishment, a number of years ago, we had a fire occurring from spontaneous combustion in a shed used for the storage of combustibles, chiefly oils and varnishes, and, in the absence of good leadership, the men attempted at first to fight the fire with water. The result was that the water flowed out over the ground, the oil floated on the water, and the fire was thus carried a considerable distance. That experience taught us to make provision whereby, in the event of fire among combustible materials which are not quenched by water, they should not be fought by using water. The remedy is a very simple one. It consists simply in making the shed or house for storing such materials with a pit two or three feet deep, and preferably with brick walls, although even wooden ones would do, the volume of the pit being sufficient to hold the contents of all the barrels and cans stored in it, so that, in the event of fire and the bursting of the barrels, the fluids would be retained in the pit until they had burnt themselves out. If, in such a case, an attempt is made to extinguish the flames, the proper material to use is not water, but sand or dirt.

Mr. F. H. Laforge.—There is one point which the author has not brought out in his paper. I have noticed, in different factories, where they have the best appliances for preventing fire, I think I can say almost universally, there are two sources of water-supply connected with the system. Am I not right, Mr. Woodbury?

Mr. Woodbury.—Yes, sir; and the necessity of two sources of water-supply to automatic sprinklers is stated in the paper.

Mr. Andrew C. Campbell.—I would like to add a few words, which I think Mr. W. F. Durfee would supply were he present.

relative to this matter of preventing serious fires where highly inflammable materials are present. Some time since a fire was started through carelessness in the japan room of the Wheeler & Wilson Manufacturing Company's factory, which resulted in the destruction of the building. Mr. Durfee was called upon as engineer to design and erect the new building, and he so arranged it that the several dipping tanks and barrels of japan are each located in a separate brick fireplace provided with a chimney. Where the fireplaces open into the room they are provided with vertically sliding fire-proof doors. The tops of the several chimneys are provided with covers to keep out rain, and so overbalanced that they tend to fly open. These covers are kept closed by an attached string which extends downward within the chimney, and so secured that, in the event of a fire in any tank, the string would burn and open the chimney top, at the same time releasing and dropping the sliding door at the base, thus allowing the fire to burn itself out without other damage. Fortunately, no fires have occurred to test the value of this method since its adoption, though no doubts exist as to its efficiency.

Mr. Jos. L. Gobeille.—I think the main cause of fire or the success of a fire is that the night-watchman, or the man in charge, does not have sufficient presence of mind or body to put it out when it starts. If we put a little more money in our night-watchmen, we would not have so many fires. Comparatively few fires occur in the day-time.

Eight or nine years ago I had charge of a wood-working establishment. There were some six or eight rooms in it. At that time the gasoline stove was a new thing, and it was supposed to be very safe. We used those stoves for heating the glue. We had perhaps fourteen or fifteen gasoline stoves of various kinds. We never had a serious fire there, although scarcely a week would pass when we did not have a dangerous-looking fire. We had a big cloth in each room, and had the men drilled so that when one of those stoves exploded they would go and put the fire out with the cloth. I noticed that in a serious fire which we did have afterward (after we got rid of the gasoline stove and put in steam and other things which were not supposed to be in anything but safe condition), it was because the night-watchman did not know enough to put the fire out when it started, but had to go quite a long way and get a policeman to show him how to get the fire-box open and turn on the alarm.

Mr. A. A. Cary.—Some mention has been made of missing fire-pails. I visited George Corliss' works a short time ago in Providence, and found that they had dispensed with all shelves, and instead had a large number of hooks on which were hung pails with a round bottom which could not conveniently be used for washing up or any other purpose than that for which they were designed.

Mr. C. J. H. Woodbury.—In reply to the questions and remarks made in the course of the discussion, I would submit, first, in reply to the first speaker, that the details of the data which he asks for in respect to the economic results of automatic sprinklers are contained in full in the paper, and were only omitted in order to comply with the brevity necessary in reading the paper. I certainly disclaim any desire to make a wrong impression relative to the liability of water damage from automatic sprinklers, and the measure of such damage is shown by the average of the whole number of accidents resulting from the presence of automatic sprinkler systems in buildings. There were, it is true, several different kinds of automatic sprinklers included in the plants on which the investigation was made, and it is conceded that there are different degrees of liability to leakage by water pressure in the various sprinklers, whatever that difference may be. The results of the investigation show that for the ground covered the amount of damage from leakage was very small. Of the 375 instances of leakage reported only 58 were due to the inability of the sprinkler to withstand water-pressure, the remaining 317 instances of leakage being due to various accidents, such as freezing, breaking of belts, and blows from machinery, mishaps in which the construction of the sprinkler bore no part whatsoever.

The relative characteristics of various automatic sprinklers in respect to sensitiveness to heat, efficient distribution of water, and resistance to leakage are in many instances apparent on examination by any good mechanic; at all events, information on these subjects is readily accessible elsewhere. It should, however, be stated that there are numerous kinds of automatic sprinklers in the market which were not installed in the property from which these facts were taken, by reason of defects of such sprinklers, principally the inability to resist continuous water pressure and also the impact of water hammer.

Some sprinklers have not been widely introduced, and there-

fore have not been exposed to as many mishaps as others; thus a division of these results among the different automatic sprinklers reported upon would have only produced a multiplicity of figures without establishing anything beyond individual distinctions.

The instance cited by another speaker respecting the failure of a fire organization appears to be an indictment against the management of that particular organization, and not against fire organizations in general; for it goes without saying that any machinery, whether it be fire apparatus or that used in the general conduct of an establishment, can be better managed by those who are practised in it and who have full and personal knowledge of all the details of the property, rather than by calling in strangers with all the rush and confusion incident to a fire alarm. The facts bear me out in this opinion, as the analysis of mill fires for a series of years shows that the large losses have not been due to a lack of what would be considered adequate apparatus, but to a lack of systematic and skilled management of that apparatus, either at the time of the fire or in keeping it in good order ready for use at such time. A mill fire-organization cannot be made merely by posting the names of a lot of men with assignments to various positions, but there must be systematic drill in the duties which may be required of them, without any of the fuss and feathers which was such a serious defect in the volunteer fire companies of years ago, and may exist to-day in town fire companies.

I must take exception to the statement of Mr. Gobeille, that few fires occur in the day-time. An experience for thirty-two years shows that three-fourths of the mill fires occur in the day-time, but that three-fourths of the losses occur in the night; thus the average night fire for that period of years was nine times as great as the average day fire, the cause of this difference being the presence of the help to extinguish the numerous fires incident to manufacturing processes occurring during working hours. During recent years, this difference between night and day fires has been largely reduced by the action of automatic sprinklers, which are of course as efficient in their action by night as by day, and also constitute an automatic fire alarm for those on the premises.

With regard to the remarks relative to the fire pails with the hemispherical bottom, it is true that such pails are not suited

for other than fire purposes, and some forms of them are not very convenient even in such instances, because the hemisphere covers the whole bottom of the pail, and the course of the water cannot be directed by the hand any more than that of a too large

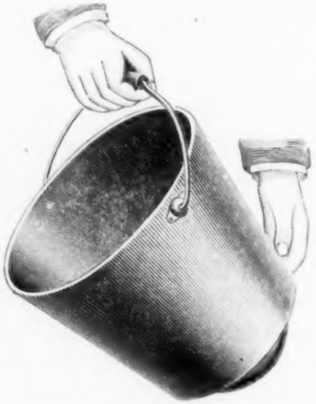


FIG. 96.

size of ninepin ball. However, in some styles of these pails the hemispherical bottom does not reach the edges, which have chines like an ordinary pail, and the course of the water thrown from the pail can in like manner be directed as in an ordinary pail (Fig. 96).

CCCLXXIII.

THE COMPARISON OF INDICATORS.

BY J. BURKITT WEBB, HOBOKEN, N. J.

(Member of the Society.)

THE question often arises as to which of two indicators is the best, and, thus stated, it may seem a simpler and more definite question than it is. To make it definite, however, it must be known for what particular use the instrument is to be "the best," for the possibility must be evident *a priori* that one indicator might be the best for one purpose and another for a different use.

Now an indicator may be used for three things:

(*a*) To obtain the area of the true card, as representing the work done per stroke,

(*b*) To obtain the pressure as some particular point of the stroke,

(*c*) To obtain the shape of the true card, as indicating the condition and action of the steam and various parts of the engine. It does not follow that, because one indicator is superior to another for one of these uses, it is so for the other two.

There are also various features in the construction of the indicator, which must be carefully examined and compared before a correct judgment of the merits of the instruments can be made. Confining what follows, for simplicity and brevity, to the ordinary forms of instrument and leaving out of consideration special forms, devised by others and by myself, the following features of construction in the indicator and its connections will affect the accuracy of the instrument in one or more of the points, (*a*) (*b*) (*c*), mentioned:

(1) Uniformity of the spring,

(2) Parallelism of the piston movement to the cylinder,

(3) Uniformity of the pencil movement,

(4) Parallelism of the pencil movement to the drum axis,

(5) Accuracy of the drum motion,

(6) Phase of the drum motion,

(7) Mass of the parts and its distribution, and the strength of the spring,

(8) Friction of the piston and pencil movements,

(9) Lost motion.

(1) A spring can be investigated and the card corrected for any lack of uniformity discovered.

(2) The parallelism of the piston movement depends mainly upon the way in which it is attached to the spring. The ideal piston should remain parallel to the cylinder throughout its motion, independent of any restraining action of the latter upon it, but the actual piston is more or less guided by the cylinder, by which action a variable friction is caused. The cylinder guides it in two ways: First, it compels the piston to slide along parallel to itself, whereas, were the cylinder absent, the piston would generally follow a different path; second, it compels the piston to remain constantly with its axis parallel to, or coinciding with, the axis of the cylinder, except in so far as the looseness of the former may allow of its tipping a little so as to make a slight angle between the axes of the piston and cylinder.

The tendency of the piston to leave the exact center of the cylinder and press against the sides of it is caused by the obliquity of the link connecting it with the pencil movement, which may generally be neglected, and by the action of the spring itself, if improperly connected.

Figures 41 to 44 illustrate such a method of connection. In Fig. 41 the spring is supposed to be under no compression and its ends are attached, at a to the cylinder cover and at b to the piston, in such a manner as to make a perfectly rigid union, such as would be made by entering the ends into holes and brazing them fast. Such a connection has the effect of fixing the angle α, at which, under all conditions of the spring, it must meet the cylinder head and piston.

Fig. 42 shows the effect of a simple compression of the spring when not confined by the walls of the cylinder, and, as steam could not be applied in this case, it may be supposed to be caused in some other way, say by making the piston b of iron, and attracting it upward by a magnet placed above it, or by pulling it up by a cord, as indicated by the force (arrow) shown. b must then stand not directly beneath a, because the compression has reduced the pitch of the spring without a corresponding diminution of the angle α, which has remained constant. Drawing then the spring

with the reduced pitch it must be tipped to one side, as shown in the figure, in order that it may meet *a* and *b* at the fixed angle α.

Fig. 43 is drawn on the supposition that the piston is now pulled to the left by the horizontal force shown until it is directly beneath *a*, and, in addition to the uniform torsion which exists throughout the wire of the spring in Fig. 42, the wire will be bent

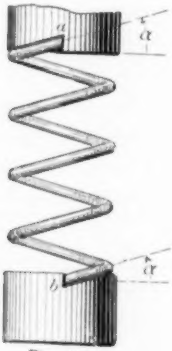


Fig. 41.

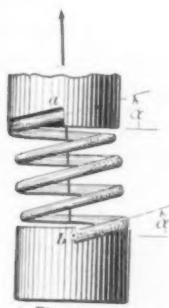


Fig. 42.

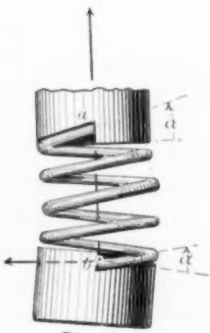


Fig. 43.

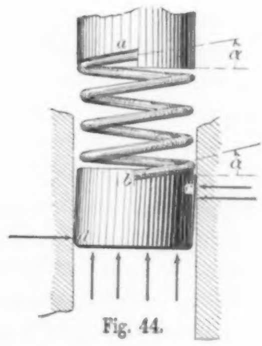


Fig. 44.

somewhat, so that the spring will assume the curved form shown. In consequence of this, the piston will lose its vertical position and incline to one side, as in the figure.

Fig. 44 shows the piston brought back again in line with *a*. To accomplish this, two equal and opposite forces, constituting a couple, have been introduced, as shown. The piston now being in position to fit the cylinder, it may be supposed to be in its

usual place therein, so that the force to compress the spring is represented in the figure as furnished by the steam beneath the piston, while the horizontal forces are furnished by the walls of the cylinder, against which the piston presses at the points *c* and *d*.

With this method of attaching the spring, therefore, the piston binds in the cylinder somewhat harder on one side than on the other, and thus causes friction, which increases as the spring is compressed. The limits of this paper prevent a discussion of duplex springs and other methods of attachment; the method discussed being chosen mainly as an illustration of the points to be considered in discussing the attachment of springs.

(3) and (4). Uniformity and parallelism of the pencil movement depend upon the geometrical nature of the pencil mechanism, and will not be discussed further than to say, that any inaccuracies in these respects can be ascertained by experiment or calculation, and corrections made therefor.

(5). The accuracy of the drum motion does not depend upon the indicator itself, but upon the method of connecting it with the engine. It is mainly a geometrical question and one but little understood, to judge from the numerous radically bad and carelessly made connections in frequent use. This subject will be passed over so far as it is a geometrical one with the statement that, like other geometrical inaccuracies, the errors resulting therefrom may be calculated or found by experiment and allowed for. The experimental determination of the error would consist in setting the piston of the engine in a number of positions and causing the indicator-pencil to describe a vertical line on the card for each position; if the positions of the piston are equidistant, the lines on the card should be the same, and such a test should be made in all cases where there is not an absolute certainty that the reducing mechanism is geometrically perfect.

When cords, however, are introduced into the reducing mechanism which actuates the drum, there is liability to an error which does not depend upon the geometrical nature of the mechanism, but upon the elasticity of the cord.

An important mechanical principle underlies the action of the drum, and the same principle affects, as will be seen later, the dynamic action of the spring, piston, and pencil connections.

It is necessary to the ideal action of the drum, that its natural speed of oscillation should agree properly with the speed of the engine, and every drum and spring have their own period of oscil-

lation. To ascertain this, the spring should be made fast to the drum, instead of being simply hooked to it, and the drum should be mounted upon centers so that it may oscillate with little friction. When the spring and drum are so proportioned to each other that they have the same time of oscillation as the engine, the motion of the drum, if there be no friction, will agree with the motion of the engine; *i. e.*, the drum will reverse at exactly the same instant as the engine does, and a cord conveying motion to the drum will remain uniformly stretched, so that there will be no change of amplitude.

The natural time of oscillation is the same whether the spring be attached firmly to the drum and the drum be free to oscillate

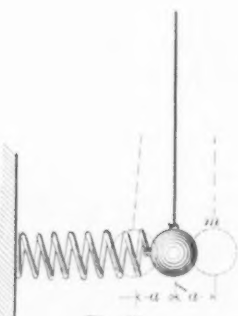


Fig. 45.

Fig. 46.

unconfined by a cord, or whether the spring be hooked to the drum and the latter drawn permanently out of its position of rest by a tightly stretched cord. To make this clearer, suppose a simple mass m, Fig. 45, attached to a spring and supported by a long string, so that it is free to oscillate horizontally uninfluenced by gravity. If the mass be now drawn a distance a to the right and set free, it will oscillate, under the action of the spring, between two positions equidistant from and on opposite sides of its central position, or position of rest, and the distance $2a$ between these two extreme positions is the *amplitude* of its oscillation. Call the time or *period* of its complete oscillation t, *i. e.*, the time required for its return to the position at which it was set free. Suppose again that the same mass be suspended from the ceiling

by the same spring, as in Fig. 46, the weight of the mass will stretch the spring as shown, and if the mass be supposed to be a unit of mass, the tension in the spring will be about 32 lbs., whereas in the central position in Fig. 45 the tension is zero. Now let the suspended mass be either raised or pulled down a distance a from its natural position and set free, it will vibrate vertically, precisely the same as it before vibrated horizontally, *i. e.*, the time t will be the same and the amplitude $2a$ the same. Suppose that in Fig. 45 the tension of the spring when extended a to the right is 10 lbs., and when compressed a to the left, -10 lbs., the minus sign indicating that the tension is compression, then at intermediate points the force of the spring will be proportionate to the distance of the mass from its central position. The same is true when the mass is suspended, except that all the tensions must have 32 lbs. added to them, so that at the extreme upper position the tension of the spring is $32 + (-10) = 22$; at the central point, $32 + 0 = 32$; at the lowest point, $32 + 10 = 42$, and at any intermediate point $32 +$ an amount proportionate to the distance of the mass below its central position, distances above being regarded as minus.

Suppose in the third place that the mass and spring are again arranged to oscillate horizontally, (see Fig. 47, in which the mass is supposed to be supported by a frictionless plane, instead of being suspended, as in Fig. 45,) and that the mass be drawn out of its central position by a constant force of 32 lbs., as in Fig. 46. Under these conditions the circumstances of the oscillation will be the same as in Fig. 46, with the exception that it will now be horizontal. In Fig. 47, b is a strong spring which will bear 32 lbs. tension with a small increase in length; d is a sliding bar with a cross-slot in which the crank c works and gives the bar a stroke $2a$ in length. If now the crank be rotated in the time t, that is, in the time in which the spring and mass naturally oscillate, then the bar and the mass may oscilate in unison, and the spring b be stretched by the constant force of 32 lbs., so that an inextensible cord might take its place. As, however, all cords are elastic, they are properly represented by the spring b, and the arrangement in Fig. 47 corresponds with an indicator-drum acted upon by its spring, and connected by a cord with the cross-head of an engine.

If the crank be revolved in a greater or less time than t, the mass will be compelled to follow it, and, therefore, to change its period of oscillation and, in consequence, a change in the amplitude

will occur. If the mass be made to oscillate in a time greater than t, the force supplied to it at each point of its path must be less than the spring s would supply, which will be the case if the spring b suitably changes its tension; thus, suppose that in the extreme left position b has shortened sufficiently to lower its tension to 31 lbs., and at the extreme right that it has lengthened so as to increase its tension to 33 lbs., while in the central position no change has occured; and suppose that the slight change in the positions of the mass, due to the change in the length of b, be neglected so far as affects the tension in s, so that the tensions in the three positions may be taken as above calculated, *i. e.*, at 22, 32, and 42 lbs.; then the force urging the mass to the right and causing the oscillation being the difference of the tensions of the springs, is $31 - 22 = 9$, $32 - 32 = 0$, and $33 - 42 = -9$ for the three principal positions and proportionate amounts for inter-

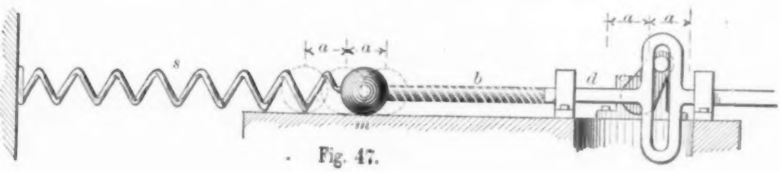


Fig. 47.

mediate ones, which force will require a time greater than t to produce an oscillation of the mass.

The effect on the amplitude will be to decrease it by an amount equal to the change of length in b, due to the change of tension from 31 to 33 lbs., which occurs during an oscillation; and in the same way, if the mass be forced to oscillate in a less time than t, the amplitude will be increased. the spring b being stretched to a greater tension than 32 lbs. at the extreme left, and to a less tension at the extreme right, so as to make the force actuating the mass greater than that required for an oscillation in the time t.

With an actual indicator and cord connection the change in amplitude will be greater or less as the cord is longer or shorter. Change of amplitude, however, is not particularly objectionable, providing the motion of the drum remains a sufficiently exact reduction of the piston motion, which has been supposed to be the case in the above treatment, so that all discussion of slight disturbances introducing harmonics has been avoided. The motion of the piston has also been supposed harmonic, as represented in Fig. 47. When the piston motion is not harmonic, as is ordinarily

the case, a long string will allow the drum motion to depart from the exact reduction of the piston motion so as to become more like the natural harmonic oscillation of the drum. With some engines having short connecting rods this would be a point requiring investigation as to the accuracy of the card if a long string connection were used.

It sometimes happens with a weak drum-spring or a heavy drum that the latter oversteps, or throws; thus, if the engine revolves in a less time than t, the force to produce oscillation in the mass m might be, for the left, center, and right positions in Fig. 47, $33 - 22 = 11$, $32 - 32 = 0$, and $31 - 42 = -11$; but if the spring s is very weak, say only one-tenth as strong as previously supposed, so that in the three positions its tensions are 2, 3, and 4 lbs. instead of 22, 32, and 42, there will be large differences to be made up by b. As the central tension of s is 3 the average tension of b must be 3, so that in the central position the force acting on the mass will be $3 - 3 = 0$; but in the extreme positions to get the 11 lbs. required, b must be stretched at the left and shortened at the right. At the extreme left b must stretch until its tension is 13, so that $13 - 2$ may $= 11$, while at the extreme right it must shorten until its tension becomes $- 7$, so that $- 7 - 4$ may $= - 11$ as required; b will then be under 7 lbs. compression, which is possible if b is a spring capable of being compressed; but if it is a cord, then as soon as the mass reaches a point to the right for which the cord tension becomes nothing, the cord will become slack and the drum will, beyond that point, oscillate under the action of s alone. This will cause the drum to overstep, inasmuch as s alone is too weak to bring it to rest at the proper point.

The only difference between the drum with its spring and the arrangement in Fig 47, is that in the latter we have a simple force urging a mass, while in calculating the action of the former the moment of the spring takes the place of the force, and the moment of inertia of the drum the place of its mass, leaving the action of the drum otherwise the same as that of a simple mass.

(6). If friction be supposed between the mass and its supporting plane, a change of phase will be produced and the oscillation of the mass will lag behind that of the sliding bar d. This is plain when it is remembered that friction involves a transmission of energy from the crank c to the mass m, and that, therefore, b must be shorter during the stroke to the left and longer during that to

the right, than it would be with no friction and the oscillation of m in unison with that of d.

The same thing occurs in other methods of transmission of energy. Suppose, Fig. 48, that a, b, and c are the cross-head, connecting-rod, and crank of a steam engine, and that d and e are the connecting-rod and cross-head of a pump driven by a crank at the farther end of the shaft, which may be of considerable length, then owing to the twisting, or torsion of the shaft, there will be a change of phase between engine and pump, so that the latter will lag behind the former.

(7). In comparing indicators, or calculating the action of any one instrument, it is not enough to consider the weight or mass of parts, as is often done, but its distribution must also be taken account of. The piston is, indeed, the only part that goes into the calculation with its simple mass, the spring and other parts go

Fig. 48.

into it with "equivalent masses," calculated in much the same way as the moment of inertia of a rotating body, and becoming the moment of inertia for any part which simply revolves. For the spring the "equivalent mass" is one-third its actual mass, to be added to the mass of the piston. This matter will not be further explained here, differing as it must with different indicators, though at some future time it may be the subject of a paper taking up one or more styles of the instrument, and calculating their action and defects.

The question of the relation of the mass of the parts to the spring is a very simple one when friction is not considered. If at the end of the stroke, or elsewhere, steam at full pressure be suddenly admitted, the piston and pencil will fly to double the height corresponding to the pressure, and continue oscillating unchecked above and below the pressure, which need not remain constant.

An alteration of the mass of the parts, or of the strength of the

spring, will cause simply a change in the time of oscillation and in the scale of the card; in fact, the state of things is the same as in Fig. 45, where a constant force acts on a mass controlled by a spring.

(8). The friction of the piston and pencil motions, and the partial throttling or friction of the steam in the passages modifies and checks the oscillations, and, after they have disappeared, it shifts the lines of the card, principally the expansion line, so as to alter the shape of the card and increase, generally, its area. The greater the friction the less will be the oscillations and the sooner they will disappear, but the greater will be the subsequent distortion of the card.

(9). Lost motion to any appreciable extent is simply imperfect construction of the instrument, and can be more or less accurately ascertained and a correction made for it.

Returning now to *a*, *b*, and *c*, the following remarks may be made as to the action of an indicator:

For the purpose *a*, the card is affected by 1, 2, on account of the friction of the piston, 3, 5, 6, 8, and 9. A fault in 2 may appreciably alter the area; faults in 1, 3, 5, and 9 may be allowed for. 6 tends to reduce the area of a steam card. 8, as far as produced by a fault in 2, generally increases it. 7 acts indirectly. The stronger the spring for the same piston and pencil movement, the less the energy stored in the oscillation at the beginning of the stroke, and the less the time required for the friction to absorb it, and the greater the remaining time during which the friction affects the area directly. During the oscillation the area is affected by the fact that the motion of the drum is not uniform. Were it so, the areas added to and taken from the true card by the oscillations might balance each other, but, because the distance of the pencil from the end of the card varies nearly as the versed line of the crank angle, oscillations near the middle of the card will occupy much more space lengthwise of the card than those at the end, and the exact balancing of the areas given and taken from the card is more complicated. The shorter the period of the oscillation the sooner it will die out, which is an additional reason for using a strong spring, the other reason being given above; so that a strong spring conduces to accuracy in *a*.

Most of the things affecting *a* affect also *b* and *c*, but in different degrees. If the pressure is desired near the end of the stroke, 7 affects it but little, and 6 has little effect on a pressure near the

beginning of the stroke. 7, independent of friction, has no effect on the shape of the card except that the oscillation make it necessary to supply a part of the card by estimating a mean line throughout the extent of the same. Friction affects both pressures and shape seriously, and considerable change of shape is often due to 5 and 6.

The drums of indicators could be made adjustable as to their time of oscillation by furnishing several springs with them. Instead of this, masses adjustable radially could be attached to the drum, by which the moment of inertia thereof could be changed. Overstepping with light springs can also be prevented by making the cord connection double, *i. e.*, by using an endless cord stretched over pulleys. I believe, however, that for accurate work it might be better to run an indicator drum continuously by a belt from the engine shaft, and to get the area of the card by adding together a sufficient number of ordinates, so spaced as to allow for the change in the card due to the change in the motion of the drum. This would simply require a suitable parallel grating for ruling the ordinates. The method would give several distinct cards in succession, instead of superimposed cards, when the operation of the pencil was not confined to one stroke, on account of the probable slight difference in speed resulting from the use of a belt instead of gearing. A planimeter might be made for such work, but planimeter work is not in general so accurate as the method by ordinates.

DISCUSSION.

Mr. H. W. Spangler.—It might be interesting, in this connection, to call attention to an article by Messrs. Reynolds and Brightmore. This is a paper on "Errors of the Indicator Diagrams" in the proceedings of The Institution of Civil Engineers of Great Britain, Vol. LXXXIII., Session 1885-86, Part I. It gives diagrams showing the effect of the inertia of the parts and the other matters which are treated of by Professor Webb in a very complete and satisfactory manner.

Mr. H. H. Suplee.—The first trial of the U. S. cruiser *Baltimore* for horse-power and speed gave rise to considerable discussion, hingeing upon this very question of the friction of the indicators with which the test was made. It seems that the friction of the instrument, or some of the instruments, had been tested to ascertain the extent of the friction, and

corresponding corrections were made in the cards. The correction was for the friction of the pencil and partially of the piston, and this measurement of the cards reduced the power below that which the contract demanded, and I think it was mainly upon this very point that the *Baltimore* went to sea on the second trial, in order to ascertain if she could make her horse-power. I do not think that details of this matter have been given to the public, but I believe they will be as soon as the report in regard to the *Baltimore* is completed. If we are measuring pressures upon a half of a square inch of area to determine the horse-power of cylinders running up to 60 or 72 inches in diameter, the importance of the error becomes correspondingly greater as the size of the engine increases.

I cannot give the exact details as to the method of allowing for variable friction. The mean height of each card was taken with the Coffin averaging instrument, and I think a mean height had previously been taken from a test card. In other words, pressures with the instrument had been compared with the actual pressures measured by other means, and these were used to show what was the error in the indicator at ten different points of the stroke. That gave a corrective amount to be added or subtracted above or below atmosphere. Corresponding ordinates were added when a card was to be measured, and the corrections were plotted inside and outside of the curve, and a new duty diagram drawn in. Then the planimeter was run around this corrected diagram. I believe some objection was made to the method of correcting the cards, because it made quite a difference in the power indicated on the cards. This shows the importance of correcting for the friction, particularly where so very large powers come into question.

Mr. Spangler.—I have been told that the correction on the cards of these tests was made substantially as Mr. Suplee has said, but practically with some slight differences. The correction card simply shows the statical difference between the pressure on the under side of the piston and the line at which the pencil stands for that pressure. These diagrams, which are nothing more than straight lines drawn and showing the pressure corresponding to a certain spring, are sent with the indicator. The indicator card is then taken and divided into two parts. One part of the card covers the admission and expansion line. The area between the zero line and the admission and

expansion line is measured and divided by the length. This gives the mean height during the forward stroke, and to this mean height is added, or from it is subtracted, the proper correction. The lower half of the card was taken in much the same way. The area below the card proper and to the line of zero pressure was measured and divided by the length, giving the mean height and the mean pressure on the back stroke. This was corrected. The difference between these two corrected dimensions was taken as the pressure. I believe that is substantially right. It has been repeated to me several times, but I do not know that it is at all correct. It may be interesting, and I give it just as I understood.

Prof. Thurston.—I would like to call attention to the article referred to still more strongly than has Professor Spangler. It presents a discussion, printed with the original paper, which covers more ground than any similar discussion with which I have ever met; and the paper, as a whole, seems a better commentary on errors in indicators, based on experimentally determined results, than anything I have ever seen.

*Prof. J. B. Webb.**—Having examined the two papers † by Professor Reynolds and Mr. Brightmore, whose complete titles, by the way, are not given in the preceding part of the discussion, some brief remarks thereon will be appropriate. Dr. Thurston's estimate of the value of the discussion upon the papers is sustained by the examination; but his characterization of them as a paper "based on experimentally determined results" does not accord with the fact that there are two distinct papers, the first of which details a "theory taught for many years in Owens College," and based mainly on the laws of mechanics, while the second records experiments just made to test the application of the theory.

I am glad to find that no essential change would have been needed in my paper had these been read before preparing it. My object was twofold; first, to put in a clear and readable

* Author's closure under the Rules.

† Paper No. 2070, "On the Theory of the Indicator and the Errors in Indicator-Diagrams," by Osborne Reynolds, M.A., LL.D., F.R.S., M. Inst. C.E.; and Paper No. 2071, "Experiments on the Steam Engine Indicator," by Arthur William Brightmore, B.Sc., Stud. Inst. C.E., late Berkeley Fellow in Owens College, Manchester. Occupying pages 1 to 20 and pages 21 to 41, with combined discussion, occupying pages 42 to 105, of the "Minutes of Proceeding of the Institution of Civil Engineers," Vol. LXXXIII.

shape for the general mechanical public the main laws of mechanics which control the action of the indicator—which can scarcely be new to any one well versed in mechanics—and, second, to call attention to some points which were supposed to be new, and which, I now find, are not mentioned in these papers.

The general method of treating the subject adopted by Professor Reynolds is much to be admired, as showing a familiarity with and reliance upon a strictly mathematical investigation. In the necessarily limited time which can be allowed and devoted to my concluding remarks in this discussion a complete review of these papers cannot be made, and it would be scarcely just, therefore, to point out as positive errors all the things which strike me as questionable. With this proviso, however, I shall feel free to question some of these points, referring to the same by pages and tenths, the decimals indicating the distance down the page.

The treatment, as a mathematical analysis of actual indicator cards, leaves much to be desired, and notably in these points: (a) In the treatment of friction; (b) in the application of the analysis to cards represented by equations, and in some degree resembling actual cards; (c) in the notation, interpretation, and discussion of the equations, with the view of exposing their anatomy and the simple laws of mechanics which they express.

In the requirements for an exact diagram, page 3.2, presumably "instantly" means in the time occupied by the change of pressure. In the causes of disturbance, page 4.4, it is difficult to imagine any advantage in enumerating the three causes as affecting the drum when but two affect the piston. On the contrary the "varying action of the spring" would seem to be a desirable thing.

Page 5.5. Equation (2) will not apply to an ordinary indicator card, even with t removed from under the radical sign.

This method, page 6.4, of estimating the centrifugal error, as it may be called, is undoubtedly elegant, if accurate; it had not previously occurred to me that it might be.

The reason stated for the "vibratory disturbance," page 7.2, seems to me to be unsupported by the equation or reason. The term (2) is not a function of the shape of the card nor of the velocity of the engine, and the statement seems to be directly contradicted by the figure and paragraph following it.

I presume that it is not intended to state, page 8.5, that the

rise of pressure could ever be instantaneous, or that the pencil could ever go to the double height. Upon this point Mr. Brightmore, page 35.1, seems to have a peculiar theory, differing from the usual one, held by Professor Reynolds. Now if any coincidence might apparently be favorable to the height of the first oscillation, it would be to have the period of the oscillation an aliquot part of that of the engine, and so little friction that the oscillations might continue unchecked throughout the card. Evidently then, also, the shorter the period of admission the greater the oscillation, but no such persistent oscillations seem to be in view.

As to the effect of friction on the diagram, pages 9.2 and 10.6, would it not be more correct to say, in the case of horizontal admission and back pressure lines, that the friction might act either to increase or decrease the diagram, according to the number of oscillations introduced by the rapid changes of pressure at the ends of the card? For an odd number of semi-oscillations the effect will be to increase the area, and for an even number it will be decreased. This may easily be seen for admission, for if the pencil be stopped by friction before the first upper semi-oscillation is completed, the pencil will remain above the admission line. While if it passes that line and is stopped during the second, or lower, semi-oscillation, it will be left beneath the line, thus reducing the area.

It seems to me that there is a slight mistake, page 9.7, where the corners are said to be cut off by 45° lines; these corners and the curved corners in Fig. 3 would seem to be common tractrices. In Fig. 4, also, on the supposition that the theoretical (dotted) card is a rectangle, with two corners replaced by the expansion and compression curves, there will be tractrices at the two rectangular corners, and at the point of cut-off the pencil line will not make an angle, as shown in Fig. 4, but will have a round corner, as shown in Fig. 104.

FIG. 104.

The treatment of the "Disturbances of the Drum," page 11.7, is far from satisfactory. It is a problem differing but little in kind from the previous one. In both there is a mass attached to a spring and having, therefore, a natural line of vibration, and an attempt is made to compel it to follow some other law of motion. In the former problem the pressure of steam, repre-

sented by a series of harmonic terms, and in this the elasticity of a cord, connecting the mass with a more or less harmonically moving point, is the compelling force. No such similarity is apparent in the mathematical treatment.

In "(3.) *The Inertia of the Drum*," page 11.9, the drum is tacitly supposed to have its spring removed and the cord kept stretched by a constant force, so that the acceleration of the drum depends upon the tension in the cord, the effect of the drum spring being taken up in the next paragraph but one. Now, in the expression yIN^2x, page 12.0, for the displacement of the drum from its true harmonic position, we have IN^2x as the accelerating force necessary at the surface of the drum and due to the change in length $(=yIN^2x)$ of the cord, whereas it would seem necessary to reduce the moment of inertia I to the radius (a, say) of the drum and to put in place of N the angular velocity of the engine.

The moment of the friction of the drum, page 13.1, could hardly cause an absolute stoppage of the same, unless it were greater than the maximum value of the moment required to oscillate it. All it could do would be to decrease the available oscillating or gyrating moment, even if it were not otherwise provided for. But, were the friction proportional to the velocity, a retardation of the phase of the drum motion would provide for it without disturbing the harmonic motion. The gyration may as well be supposed harmonic; if a short connecting-rod be introduced, as in the figure, page 13.5, its effect on the harmonic motion should be discussed mathematically in another place, inasmuch as it has nothing to do with the question of stoppage at the ends of the diagram.

The effect of uniform or slightly varying friction, as differing from friction supposed to vary with the velocity, will be substantially the same as regards change of phase, with the addition of a disturbance of the exact harmonic gyration sufficient to equalize the force devoted to overcoming the friction. The drum acts, in fact, as a fly-wheel, absorbing the excess of energy furnished near the middle of the card by the change of phase and giving it out near the ends. It follows, also, that the distance of the drum behind its true position can hardly be constant, so that the analysis on page 14 is of doubtful accuracy.

This drum friction is, of course, variable, increasing from one end of the card to the other.

I can scarcely imagine that (page 15.0) any one with a fair knowledge of analytical and practical mechanics could carefully consider the action of an indicator drum without perceiving the effect of friction. I had, in fact, supposed it to be a phenomenon familiar to experts upon indicator cards, who may find but little to help them in the rest of the paper, especially as the applications are not to the modern forms of instruments now exclusively used in good practice.

The apparatus employed, page 39.5, to detect the effect of friction is admirable and the results interesting. I cannot see in them any evidence that the drum remains at rest at the ends of the card.

Would it not be near one end of the card that the cord might become loose, page 41.6, rather than near the middle?

Why M_m, page 38.5, should be constant I cannot see, nor why such a complicated law for the friction moment M_f is supposed. The statement that "the tension T is a maximum," etc., page 38.7, cannot be correct; when the time of oscillation of the drum agrees with that of the cross-head of the engine the tension T would be constant, except for the friction. Then the drum spring operates the drum without any help from the tension T. At other speeds of the engine the drum spring is assisted more or less by the elasticity, not by the mean tension, of the cord. As a means of operating the drum the drum spring seems to be ignored, as if its only office were to keep the cord stretched. On page 36 the weight of the parts is supposed to affect the finding of the pressure by drawing a line midway between the crests and hollows of the oscillations, and this weight comes into the equation, page 36.7, from page 28, where it has been put into the value of Q. How it can affect this value I cannot see, because pressures are reckoned from an atmospheric line drawn on the card, by which means the weight is eliminated, whether the instrument be used upright or inverted. The pressure line is evidently such an average where the pressure is constant and there is no friction, as is here supposed; the more interesting case is where the pressure varies and there is friction. By the way, also, page 28.9, the spring should not enter the equation for the value of W' with half its weight.

What is wanted is a mathematical treatment of the indicator, with friction included, and with deductions drawn logically therefrom, but excluding doubtful speculations.

CCCLXXIV.

ON THE INFLUENCE OF THE STEAM-JACKETS ON THE PAWTUCKET PUMPING ENGINE.

BY J. E. DENTON, HOBOKEN, N. J.
(Member of the Society.)

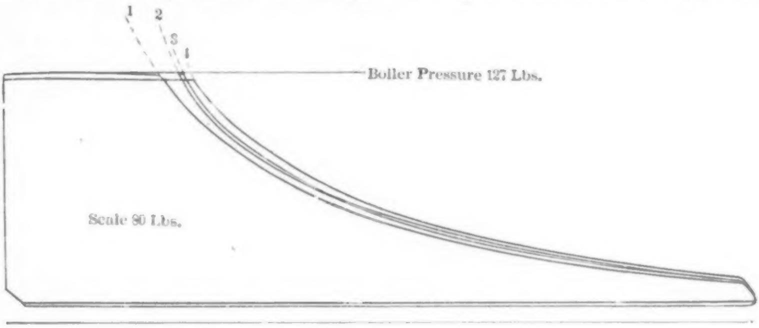
THE engine under notice is of the double expansion type, cylinder 15 × 30½ bore, and 30 inches stroke, cranks at right angles, cylinders connected with a receiver of four times the capacity of high cylinder; average revolution, 49.5 per minute; average boiler pressure, 125-lbs. gauge; average cut-off in high cylinder, one-fourth, and one-third in low.

Jackets envelop the heads and barrels of both cylinders, and steam of full boiler pressure is used in each. The condensed steam from the jackets is pumped into the feed-pipe at a point between the boiler and hot well. The condensed steam collected in the receiver is received in a trap, and continuously pumped through a heater placed in the chimney flue, and thence returned to the top of the receiver. Out of a total of 155 lbs. thus circulated, one-third only is evaporated and returned to the receiver as steam; the other two-thirds gradually accumulate in the receiver and is blown to waste every three hours.

Results of tests* of this engine are given in Table I. By the 2d and 3d tests in this table it will be seen that when the engine was in use with steam in the jackets, 107 lbs. of steam condensed per hour in the jackets of both cylinders together, and that when the total steam consumed per hour was 2,046 lbs., the indicated horse-power was nearly 148.

* The tests of this engine have been made by the writer as a matter of general interest, as its performance has been extraordinary ever since its erection by Mr. Corliss in 1878. During the year ending November 30, 1888, the Water Works report claimed an annual duty of 123,656,000, exceeding its previous performances by several million foot pounds. As no determination of the *steam consumption and boiler evaporation had ever been published*, the determinations of duty depending on measurements of coal and water pressure only, the writer requested permission to make these measurements and to verify the extraordinary duty of last year. Such permission was cordially granted by the superintendent, Mr. Edwin Darling, who, together with the chief engineer, Mr. John Walker, co-operated with the writer in every possible way to make the investigation com-

Also when steam was shut off from both jackets, and the engine made to perform as nearly as possible the same work, the consumption per hour was 2,070 lbs. This would indicate a slight gain by the use of the jackets, which would amount to about one-sixth of a pound of steam per hour per indicated horse-power.



No. 1, 72 Hour Test. Cut off 0.220 } With Jackets.
" 2, 24 " " " " 0.235 }
" 3 6 " " " " 0.242 } Without Jackets.
" 4, 9 " " " " 0.253 }

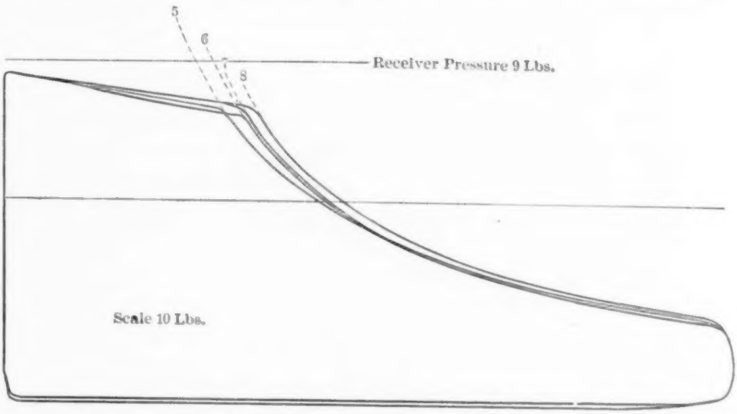
FIG. 49.

It appears, however, that the variation in indicator cards, taken under the most perfectly invariable conditions of action available in practice, is such that the mean effective pressure of the cards in Fig. 49 may be in error one pound per square inch, and that of Fig. 50 one-quarter pound per square inch, so that the indicated horse-power may be in error three * per cent. Also, that the cut-off determined from cards may err one-half of one per cent. These facts make it possible for the steam per hour per *indicated* horse-power to be indeterminate within $\frac{4}{10}$ lbs. in 13 lbs., and for 90 lbs. of steam out of 2,000 to be due to differences of cut-off not shown by the average cards in comparative tests with and without jacket. It would be quite permissible, therefore, so to vary the

plete. Funds for the expenses of the tests were generously subscribed by the trustees of the Stevens Institute, Joel Sharp, A. C. Humphreys, and Messrs. H. C. White, A. P. Trautwein, W. W. Dashiel, and the Stevens Institute Alumni Association. The results obtained fully sustain the Water Works report of performance. The present paper is abstracted from the full report, not yet published, but a digest of which was presented to the Fall River meeting of the New England Water Works Association.

* It should be understood that this error does not enter the calculation of the duty of the engine, which depends on the measurement of coal and water pressure against the plungers, both of which quantities are determinable to within $\frac{1}{4}$ of one per cent.

data of horse-power and steam in cols. (12) and (13) of Table I., as to make the consumption per horse-power in col. (14) be in favor of "no jacket." This has been the result in some cases of tests of engines. The pumping engine at Wilmington, Del., for example, gave, on comparative 24-hour tests, 19.8 lbs. steam per hour per horse-power with jackets (including consumption of latter), and 17.9 lbs. without jacket, the latter being confined to



No. 5, 72 Hour Test. Cut off 0.310 } With Jackets.
" 6, 24 " " " " 0.3345 }
" 7, 6 " " " " 0.335 } Without Jackets.
" 8, 9 " " " " 0.350 }

FIG. 50.

the high cylinder. The difference of two pounds per hour per horse-power represents *seven times the jacket consumption per hour.*

I believe such results to be due to errors in securing, or in reducing results to, truly identical conditions with and without jacket. For it is improbable that there should be more than a very slight loss of economy from the jacket for the following reasons:

Referring to column (15) Table I., it is seen that 460 lbs. of steam is condensed during admission. Thereby 68 thermal units per stroke are absorbed by the metal surfaces of some part of the interior of the cylinder. Again by reference to Fig. 51 it will be seen that there is re-evaporation, represented by the distance $d_3 + d'_3$. This amount of re-evaporation requires 17 thermal units per stroke to emanate from the cylinder walls, and the increase of mean effective pressure thereby is a considerable frac-

tion of the whole card. Treating the cylinders as well-covered pipes, the loss of heat by radiation is about 20 lbs. of steam per hour, thus leaving 87 lbs. of steam per hour to be consumed in the jackets by heat taken from them by the steam in the cylin-

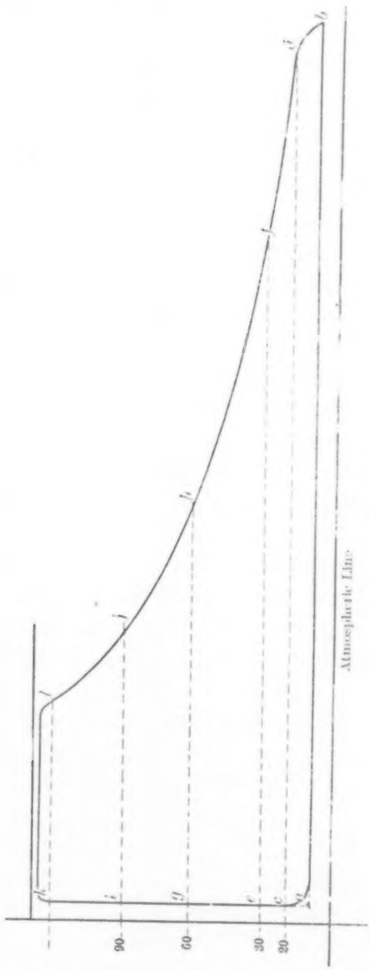


FIG. 51.

ders. Now, 87 lbs. of steam per hour represents about 13 thermal units per stroke, which is one-fifth of the heat causing the expense of 460 lbs. during admission, and almost equals the heat causing the considerable increase of card area during expansion in the high cylinder. It is not reasonable, therefore, to

suppose that this heat does not make some practical difference in the average temperature of the cylinder, which must have value in either increasing the area of the card during expansion, or reducing cylinder condensation during admission to the cylinders, or both. So far as the cards available are concerned, there is no difference in the law of the expansion curve with and without jackets, as shown in Fig. 51, although less variations in length, friction, etc., in the cards, may lead to a different conclusion later. The influence of the jacket heat must therefore be confined to the reduction of cylinder condensation during admission, and it is improbable that it does not save an amount of steam equal to that expended in the jacket less radiation, while it is quite possible that the reduction of cylinder condensation may be slightly greater in amount than the expense in the jacket. Excessive claims of saving by the jacket amounting to a multiple of the amount of steam used by it, are, however, difficult to conceive, if no difference of temperature or pressure exists between the steam in the jacket and that in the cylinder during admission. In Table II. is given the water per hour per horse-power as calculated from the cards. It averages about $9\frac{3}{4}$ lbs., and does not essentially differ between cut-off in the high cylinder and release in the low cylinder.

There is re-evaporation in both cylinders, about 8 per cent. in the high and 6 per cent. in the low. The expansion lines agree most nearly with the Mariotte law. The engine was carefully tested regarding leakage of the valves and pistons. The greatest leakage was at one admission valve on the low cylinder, and was but 5 lbs. of steam per hour, or $\frac{1}{4}$ of one per cent.

Appended is a description of the manner of plotting the points of the different cards in Fig. 70.

CONCLUSIONS:

1. That the averages of results of indicator cards taken in the most careful manner with the best modern indicators, show a possible saving from the use of jackets amounting to from 0.13 to 0.35 lbs. of steam per hour per horse-power, but that these amounts are within the limit of error to which the determination of indicated horse-power and cut-offs are subjected, so that

2. The most that can be claimed for the jacket is that it probably caused no loss, and may possibly have caused a saving, not exceeding 3 per cent. of the total steam consumption.

TABLE I.

STEAM CONSUMED BY JACKETS....107 POUNDS PER HOUR.

CLEARANCE OF HIGH PRESSURE CYLINDER = 4 PER CENT. OF PISTON DISPLACEMENT.

CLEARANCE OF LOW PRESSURE CYLINDER = 3.7 PER CENT. OF PISTON DISPLACEMENT.

RATIO OF VOLUME OF CYLINDERS = 4.085.

Duration of Test. Hours.	Condition of test.	Boiler pressure gauge. Lbs. per sq. inch.			Water pressure against pump piston, including suction. Lbs. per sq. inch.			Revolutions per minute.	Average cut-off in percentage of stroke, clearance not included.		Indicated horse-power.	Steam consumption. Lbs. per hour. Actual measurement.		Steam condensed, calculated from cards.		Mean effective pressure. Lbs. per sq. inch.		Vacuum.	Average back pressure.
		Average.	Minimum.	Maximum.	Average.	Minimum.	Maximum.		High Cylinder.	Low Cylinder.	I. H. P.	Total.	Per I. H. P.	In cylinder during admission. Lbs. per hr.	Per cent. of total feed water.	High Cylinder.	Low Cylinder.	Inches of Mercury.	The pressure. Inches.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
72	With jacket....	127	125	130	115	114	116½	49.03	0.220	0.3100	143.49	1958	13.62†	453	23*	59.5	12.60	28½	1.25
24	With jacket....	128	125	130	116	115	117½	49.50	0.235	0.3355	147.95	2046	13.82†	460	22½*	61.8	12.65	28½	1.25
6	Without jacket.	125	124	127	116½	115½	117	40.40	0.246	0.3350	148.37	2070	13.95	586	28	62.0	12.70	28½	1.25
9	Without jacket.	124	120	127	118	117	120½	48.95	0.258	0.3500	149.74	2122	14.17	551	26	63.5	12.85	28¾	1.25

Av. Barometer, 30.00 ins. * The 5 per cent. of steam condensed by jackets is not included in these figures.
† The 5 per cent. of steam condensed by jackets is included in these figures.

TABLE II.

STEAM CONSUMPTION PER INDICATED HORSE-POWER, CALCULATED FROM INDICATOR CARDS.

Cylinder.	Pressure in lbs. per sq. inch. Absolute.		Percentages of stroke (including clearance).						Theoretical water consumption. Lbs. per hour per H. P.		Mean effective pressure.				Conditions of tests.
			High Cylinder.				Low Cylinder.				High Cylinder.		Low Cyl'der.		
			Near Cut-off.		Near Release.										
	Near Cut-off.	Near Re-lease.	Actual.	Actual, divided by ratio of cylinders.	Actual.	Actual, divided by ratio of cylinders.	Near Cut-off. Actual.	Near Re-lease. Actual.	Near Cut-off.	Near Release.	Actual.	Actual, divided by ratio of cylinders.	Actual.	Total mean pressures reduced to low cylinder. Sum of Columns 13 and 14.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	
High Cylinder....	134.7	39.7	25.99	6.37	90.00	22.06	....	....	9.86	10.87	59.5	14.58	......	27.18	With Jacket.
Low Cylinder ...	19.7	7.7	....	....	....	....	34.0	93.7	8.78	9.76	....	....	12.60		72-hour Test.
High Cylinder....	134.7	39.7	29.01	7.11	96.49	23.65	....	....	10.53	11.08	63.5	15.56	......	28.41	Without Jacket.
Low Cylinder	19.7	7.7	....	....	....	....	38.7	99.9	9.36	9.96	....	....	12.85		9-hour Test.

APPENDIX I.

EXPLAINING METHOD OF PLOTTING POINTS IN EXPANSION LINES IN FIG. 51 AND PLATE, FIG. 70.

The actual cards as per Figs. **51** and 69 have been measured as follows, referring to Table **III.**:

Column 2 gives the total length, *ab*, of the card in inches; columns 3, 7, 11, 15, 19, 25, 29, and 33, give the actual lengths of

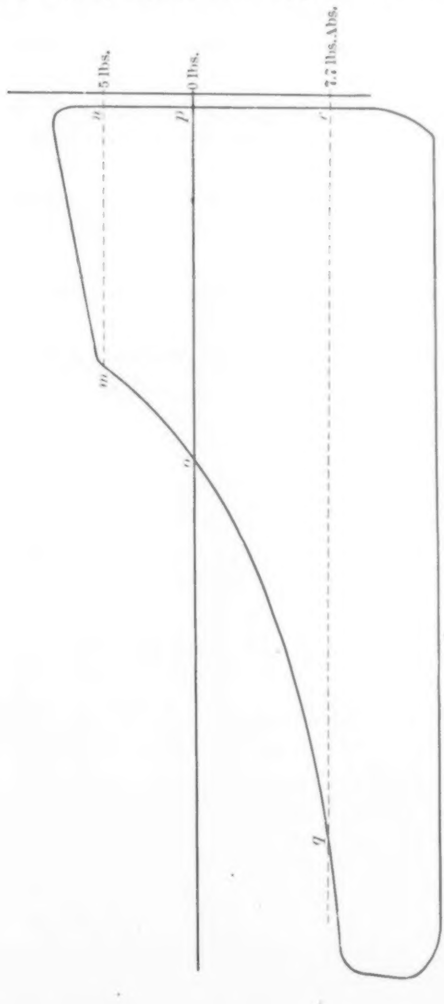
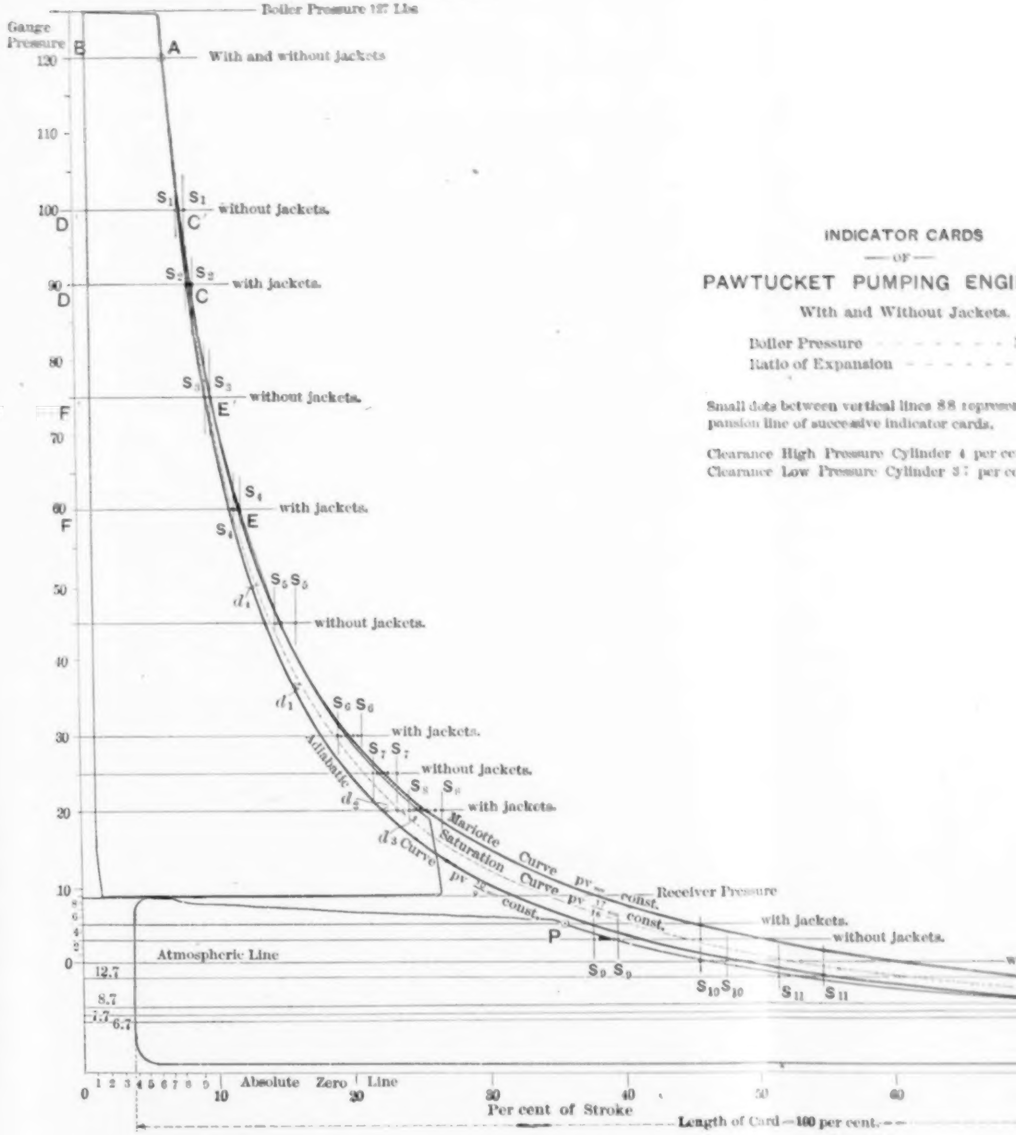


FIG. 69.

line *kl, ij, gh, ef, cd, mn, op, qr;* columns 4, 8, 26, 30, etc., give the same distances with the clearance added, namely, 4 per cent. of *al* for the high pressure cards, and 3.7 per cent. for the low pressure cards; columns 5, 9, 13, 17, and 21, give distances in the preceding columns, expressed in per cent., of *ab*, and divided by the ratio of the volumes of the cylinders, namely, 4.08; columns 27, 31, and 35, give the same ratios without dividing by the ratio of the cylinders; columns 6, 10, 28, 32, 36, give values in preceding columns reduced to the length of card No. 99 as unity for the high pressure cards, and to No. 34 for the low pressure cards. These cards are shown in the plate to the same scales of the horizontal scale representing volumes, the length of the low pressure card being 100.

The distance AB represents the cut-off, common to all the high pressure cards, and the point P is common to all low pressure cards. The values in columns 10, 14, 18, etc., are laid off along the horizontal lines CD, EF, etc., for cards taken with jackets, and along $C'D'$, $E'F'$, etc., for cards taken without jackets. A curve drawn through the point A, and any point at C or E or $C'E'$, belonging to the same card, forms the curve of expansion for that card. The vertical lines SS, S_1S_2, etc., show the limits between which fall the points in the expansion lines of each of the twelve cards taken under the two conditions in each cylinder and incorporated in Table III.



INDICATOR CARDS

—OF—

PAWTUCKET PUMPING ENGI

With and Without Jackets.

Boiler Pressure

Ratio of Expansion

Small dots between vertical lines SS represent pansion line of successive indicator cards.

Clearance High Pressure Cylinder 4 per ce
Clearance Low Pressure Cylinder 3.7 per ce

FIG. 70.

J. E. DENTON.

os

ENGINE No. 1.

ckets.

– – 127 Lbs.

– – – 16 to 1

epresent points in ex-
rds.

per cent.

per cent.

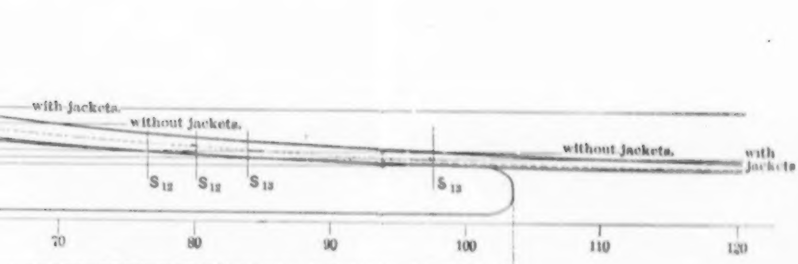


TABLE III.

TABLE OF PERCENTAGES OF VOLUME FOR VARIOUS PRESSURES.

Cards from Test with Jackets in use.

HIGH PRESSURE CYLINDER.

		Volumes for the Pressures.																			
		120 lbs.				90 lbs.				60 lbs.				30 lbs.				20 lbs.			
				Per cent.				Per cent.				Per cent.				Per cent.				Per cent.	
Card number.	Total length. l	Actual.	$+ 0.4 \times l$.	Actual.	Assuming card 99 as unity.	Actual.	$+ .04 \times l$.	Actual.	Assuming card 99 as unity.	Actual.	$+ .04 \times l$.	Actual.	Assuming card 99 as unity.	Actual.	$+ .04 \times l$.	Actual.	Assuming card 99 as unity.	Actual.	$+ .04 \times l$.	Actual.	Assuming card 99 as unity.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
Head End.																					
13.....	4.79	1.06	1.25	6.39	6.56	1.40	1.59	8.12	8.34	2.03	2.22	11.40	11.65	3.49	3.68	18.81	19.31	4.68	4.71	24.07	24.72
88.....	4.80	1.11	1.30	6.63	6.56	1.44	1.63	8.31	8.22	2.12	2.31	11.78	11.65	3.73	4.92	20.00	19.78	4.59	4.78	24.38	24.11
89.....	4.80	1.08	1.27	6.48	6.55	1.42	1.61	8.21	8.31	2.07	2.26	11.53	11.67	3.54	3.73	19.02	19.25	4.48	4.67	23.82	24.11
99.....	4.81	1.10	1.29	6.56	6.56	1.47	1.66	8.48	8.48	2.17	2.36	12.01	12.01	3.69	3.88	19.75	19.75	4.68	4.87	24.68	24.68
105.....	4.80	1.13	1.32	6.73	6.56	1 47	1.66	8.46	11.88	2.20	2.39	12.19	11.88	3.87	3.87	19.74	19.24	4 68	4.87	24.84	24.21
Crank End.																					
24.....	4.84	1.06	1.25	6.32	6.56	1.40	1.59	8.04	8.35	2.01	2.20	11.13	11.55	3.59	3.78	19.12	19.85	4.69	4.88	24.68	25.61
101.....	4.86	1.09	1.28	6.45	6.56	1.46	1.65	8.32	8.46	2.08	2.27	11.44	11 63	3.74	3.93	19.81	20.15	4.71	4.90	24.70	25.11
106.....	4.81	1.06	1.25	6.36	6.56	1.44	1.63	8.30	8.56	2.04	2.23	11.34	11.70	3.77	3.96	20.50	20.76	4.72	4.91	24.94	26.65
115.....	4.90	1.05	1.25	6.24	6.56	1.40	1.69	7.99	8.40	2.02	2.22	11.09	11.65	3.57	3.77	18.84	19.80	4.73	4.93	24.63	25.89
143.....	4.77	1.04	1.23	6.31	6.56	1.40	1.59	8.16	8.49	2.03	2.22	11.39	11.85	3.36	3.55	18.22	18.95	4.51	4.70	24.12	25.08

LOW PRESSURE CYLINDER.

		Volumes for the Pressures.											
		5 lbs. Gauge.				0 lbs. Gauge.				7.7 lbs. Absolute			
				Per cent.				Per cent.				Per cent.	
Card number.	Total length. l	Actual.	$+ .037 \times l$.	Actual.	Assuming average at 5 lbs. unity.	Actual.	$+ .037 \times l$.	Actual.	Assuming average at 5 lbs. unity.	Actual.	$+ .037 \times l$.	Actual.	Assuming average at 5 lbs. unity.
23	24	25	26	27	28	29	30	31	32	33	34	35	36
Head End.													
26.....	4.71	1.42	1.59	33.47	35.47	1.92	2.09	44.37	46.59	3.86	4.03	85.56	90.74
58.....	4.71	1.45	1.62	34.43	35.47	1.95	2.12	45.01	47.26	3.92	4.09	86.84	91.18
97.....	4.64	1.48	1.65	35.56	35.47	2.00	2.17	46.77	46.74	3.94	4.11	88.58	88.31
121.....	4.70	1.47	1.64	34.90	35.47	1.99	2.16	45.96	46.69	3.92	4 09	87.02	88.41
128.....	4.68	1.51	1.68	35.90	35.47	1.99	2.16	46.15	45.61	3.98	4.15	88.67	87.61
Crank End.													
34.....	4.74	1.38	1.55	32.70	35.47	1.92	2.09	44.09	47.62	3.99	4.16	87.76	94.78
61.....	4.65	1.34	1.51	32.47	35.47	1.86	2.03	43.66	47.59	3.84	4.01	86.24	94.01
101.....	4.63	1.44	1.61	34.78	35.47	1.98	2.15	46.44	47.32	4.26	4.43	95.68	97.50
129.....	4.61	1.39	1.56	33.84	35.47	1.91	2.08	45.12	47.29	4.13	4.30	93.28	97.76
146.....	4.64	1.36	1.53	32.98	35.47	1.88	2.05	44.18	47.49	3.46	3 63	78.23	84.10

Cards from Test with no Jackets in use.

HIGH PRESSURE CYLINDER.

Card number.	l Total length.	120 lbs. Gauge. Actual.	120 lbs. Gauge. + .04 × l.	120 lbs. Gauge. Per cent. Actual.	120 lbs. Gauge. Per cent. Assuming card 99 as unity.	100 lbs. Gauge. Actual.	100 lbs. Gauge. + .04 × l.	100 lbs. Gauge. Per cent. Actual.	100 lbs. Gauge. Per cent. Assuming card 99 as unity.	75 lbs. Actual.	75 lbs. + .04 × l.	75 lbs. Per cent. Actual.	75 lbs. Per cent. Assuming card 99 as unity.	45 lbs. Actual.	45 lbs. + .04 × l.	45 lbs. Per cent. Actual.	45 lbs. Per cent. Assuming card 99 as unity.	25 lbs. Actual.	25 lbs. + .04 × l.	25 lbs. Per cent. Actual.	25 lbs. Per cent. Assuming card 99 as unity.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
Head End.																					
.....	4.92	1.19	1.39	6.91	6.56	1.42	1.62	8.06	7.65	1.89	2.09	10.39	9.87	2.94	3.14	15.62	14.84	4.51	4.71	23.43	22.35
.....	4.84	1.15	1.34	6.78	6.56	1.34	1.53	7.74	7.51	1.81	2.00	10.12	9.79	2.83	3.02	15.29	14.80	4.29	4.48	22.67	22.01
.....	4.86	1.14	1.33	6.70	6.56	1.35	1.54	7.76	7.60	1.82	2.01	10.13	9.92	2.86	3.05	15.36	15.05	4.36	4.55	22.93	22.47
3 '89 M .52	4.89	1.26	1.46	6.30	6.56	1.47	1.67	8.35	8.71	1.93	2.13	10.65	11.09	3.01	3.21	16.06	16.71	4.45	4.65	23.25	24.20
M. .25	4.88	1.26	1.45	7.28	6.56	1.49	1.68	8.43	7.79	1.96	2.15	10.79	9.72	3.06	3.25	16.32	14.70	4.58	4.76	23.93	22.56
23....	4.88	1.28	1.47	7.37	6.56	1 51	1.70	8.53	7.59	1.98	2.17	10.89	9.69	3.07	3.26	16.36	16.03	4.57	4.76	23.88	23.35
Crank End.																					
6.....	4.88	1.20	1.39	6.97	6.56	1.43	1.62	8.13	7.65	1.85	2.04	10.23	9.63	2.89	3.08	15.45	14.53	4.53	4.72	23.68	22.28
7.....	4.85	1.16	1.35	6.82	6.56	1.38	1.57	7.92	7.62	1.81	2.00	10.10	9.72	2.80	2.99	15.29	14.71	4.38	4.57	23.08	22.20
9.....	4.88	1.19	1.38	6.92	6.56	1.39	1.58	7.93	7 51	1.86	2.05	10.28	9.74	2.88	3.07	15.40	14.59	4.44	4.63	23.23	22.02
ne 3 . M. .30	4.89	1.27	1.47	7.35	6.56	1 57	1.71	8.55	7.62	1.95	2.15	10.75	9.60	3.03	3.23	16.15	14.41	4.73	4.93	24.66	22.00
. M. .15	4.90	1.28	1.48	7.39	6.56	1.53	1.73	8.61	7.67	[illegible]	2.18	10.89	9.67	3.10	3.30	16.48	14.63	4.73	4.93	24.63	21.87
57....	4.88	1.25	1.44	7.22	6.56	1.48	1.67	8.38	7.61	1.95	2.14	10.73	9.76	2.97	3.16	15.86	14.41	4.68	4.87	24.43	22.20

LOW PRESSURE CYLINDER.

Card number.	l Total length.	3 lbs. Gauge. Actual.	3 lbs. Gauge. + .037 × l.	3 lbs. Gauge. Per cent. Actual.	3 lbs. Gauge. Per cent. Assuming average at 5 lbs. unity.	12.7 lbs. Absolute. Actual.	12.7 lbs. Absolute. + .037 × l.	12.7 lbs. Absolute. Per cent. Actual.	12.7 lbs. Absolute. Per cent. Assuming average at 5 lbs. unity.	8.7 lbs. Absolute. Actual.	8.7 lbs. Absolute. + .037 × l.	8.7 lbs. Absolute. Per cent. Actual.	8.7 lbs. Absolute. Per cent. Assuming average at 5 lbs. unity.
23	24	25	26	27	28	29	30	31	32	33	34	35	36
Head End.													
182.....	4.68	1.64	1.81	38.68	38.91	2.32	2.49	53.21	53.53	3.50	3.67	78.42	78.89
192.....	4.71	1.63	1.80	38.21	38.48	2.30	2.47	52.44	52.81	3.48	3.65	77.49	78.03
194.....	4.68	1.65	1.82	38.21	38.89	2.33	2.50	53.42	53.42	3.49	3.66	78.21	78.21
J'e 3 '89 A. M. 11.29	4.68	1.67	1.84	39.32	39.32	2.35	2.52	53.84	53.84	3.57	3.74	79.91	79.91
11.50..	4.67	1.59	1.76	37.69	37.69	2.34	2.51	53.75	53.75	3.49	3.66	78.37	78.37
P. M. 3.40..	4.61	1.68	1.85	40.12	39.52	2.40	2.57	55.75	54.91	3.64	3.81	82.65	81.38
Crank End.													
184...	4.61	1.82	1.99	43.16	38.58	2.56	2.73	59.22	52.88	3.97	4.14	89.80	80.19
188...	4.63	1.67	1.84	39.74	36.00	2.40	2 57	55.50	50.28	3.76	3.93	84.88	76.90
189.....	4.68	1.69	1.86	39.75	38.80	2.25	2.52	53.80	52 51	3.58	3.75	80.13	78.21
June 3 A. M. 11.37.	4.70	1.78	1.95	41.48	39.28	2.49	2.66	56.56	53.56	3.80	3.97	84.47	79.99
11.47...	4.71	1.74	1.91	40.55	38.93	2.44	2.61	55.42	53.20	3.78	3.95	83.86	80.50
P. M. 3.45.	4.68	1.83	2.00	42.73	38.31	2.55	2.72	58.12	51.84	3.88	4.05	86.54	77.19

Ratio of Volume of High to Low Pressure Cylinder.......................4.085

APPENDIX II.

ABSTRACT OF EXPERIMENTS PERFORMED.*

Between May 30 and June 1, 1889, the engine was run continuously seventy-two hours under its usual working conditions and measurements made of the coal consumed, anthracite stove size being used; the horse-power performed, both for pumping the water and to overcome the friction of the engine itself; the pounds of steam generated in the boilers and used to operate the engine; the pounds of water used to condense the steam as it left the engine; all temperatures and detailed measurements necessary to determine the heat rejected by the engine; the steam condensed in the jackets; the economy due to passing the condensed steam from the receiver through heater in chimney flue, and the dryness of the steam.

The duty of the plant, the steam per hour per horse-power, and the evaporation of the boilers per pound of coal were the principal elements of the quantities sought.

Between June 1st and June 2d the engine was run twenty-four hours under the same conditions as before, but with selected Georges Creek bituminous coal for fuel.

The same determinations were made, but the special object was to provide a check upon the first test and to ascertain how much gain in duty could be expected from choice bituminous coal as compared to the stove anthracite used in the first test and in the usual operation of the plant.

Between June 3 and June 6, 1889, short tests of from six to nine hours were made, the engine being sometimes run with, and sometimes run without, jackets. No measurements of fuel were made, the object sought being to determine the steam consumed by the engine per horse-power, with and without jackets in action.

PREPARATIONS FOR THE TESTS.

These consisted of the following:

The insertion of a water meter in the line of feed-pipe delivering water from the hot well to the boilers.

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* For full details see *Engineering News*, December 28, 1889, *et seq.*

The insertion of a group of three water meters in the line of exhaust pipe leading from the hot well to the river, into which the water wasted.

The insertion of thermometers into mercury-filled tubes, the latter being screwed into the steam and water pipes at various places.

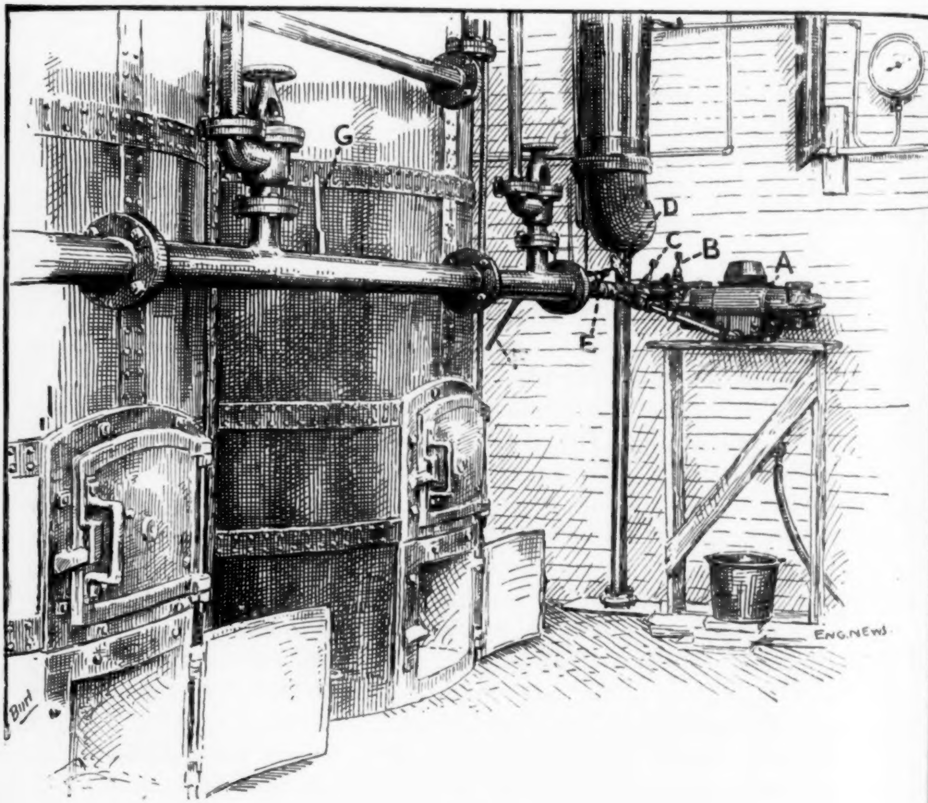

FIG. 99.
BOILER ROOM WITH METER ATTACHED TO FEED-PIPE.

The testing and sealing of the scales upon which the coal was weighed.

Means for securing indicator cards, recording the revolutions, testing pressure gauges, etc., of the most satisfactory character were already at the station.

The arrangement of the feed-water meter is shown in Fig.

99. A is meter arranged between two branch pipes, which lead out of the main feed-pipe at right angles to the latter through the two valves, C and B. Between the latter is another valve, D, lying in the line of the main feed-pipe.

By closing valve D and opening C and B the feed-water is forced to travel through B and the meter, A, and regain the main feed-pipe through C. Hence it passes into the boiler through the vertical valves leading upward out of the horizontal pipe running along the front of the boilers. G is the thermometer which measures the temperatures of the feed-water as it enters the boilers. Should any accident befall the meter, valves C and B could be instantly closed and D opened, thereby returning the feed system to its normal condition.

VALUE AND TEST OF METER.

The use of a meter to measure the feed-water has the great advantage of disturbing none of the conditions of regular working. If the hot well water is run into weighing tanks, a separate pump is necessary to feed the boilers, and in handling the steam to run the pump the temperature of the feed-water and quantity of steam generated per minute of time, etc., are altered from the normal conditions.

The Worthington meter, once rated, does not change its rate with clean water within any interval such as is consumed by tests, and it is much more satisfactory for night and day continuous work than any human labor which could be assigned to the weighing out of tanks. The rating of the meter in the present case was performed as follows :

Valve E was closed and a valve not shown opened, so that the water passing the meter all escaped by a nipple, F, which led off from the feed-pipe between the valves E and D. From F a hose was led to four barrels holding about $5\frac{1}{2}$ cubic feet each. The first few seconds' flow from F was led into a pail until the meter read an integral number of cubic feet. It was then switched into the first barrel, and by watching the increase of weight each ten seconds, the valve F could be adjusted in thirty seconds, so that water flowed at the same rate as when feeding the boilers.

From ten to twenty cubic feet, as shown by the meter, were then collected in the barrels and weighed, thus rating the meter under the exact conditions of current which existed while the boilers were being fed.

The following table gives the results of such tests:

Date.	Volume Registered by Meter.	Weight per cubic feet in lbs.
May 27, 1889.	10 cubic feet..................................	66.25
" 31, "	20 cubic feet, first 5 cubic feet................	66.60
" " "	" " " second 5 cubic feet.............	66.04
" " "	" " " third 5 cubic feet..............	66.60
" " "	" " " fourth 5 cubic feet.............	66.40
	Average of all four barrels..................	66.40
	Total weight................................	66.31
June 1, 1889.	10 cubic feet................................	66.40
	Grand average....	66.375

The group of meters for measuring the hot well are shown in Fig. 100 at *B*, *C* and *D*. Two are 2-inch Crown meters and the one

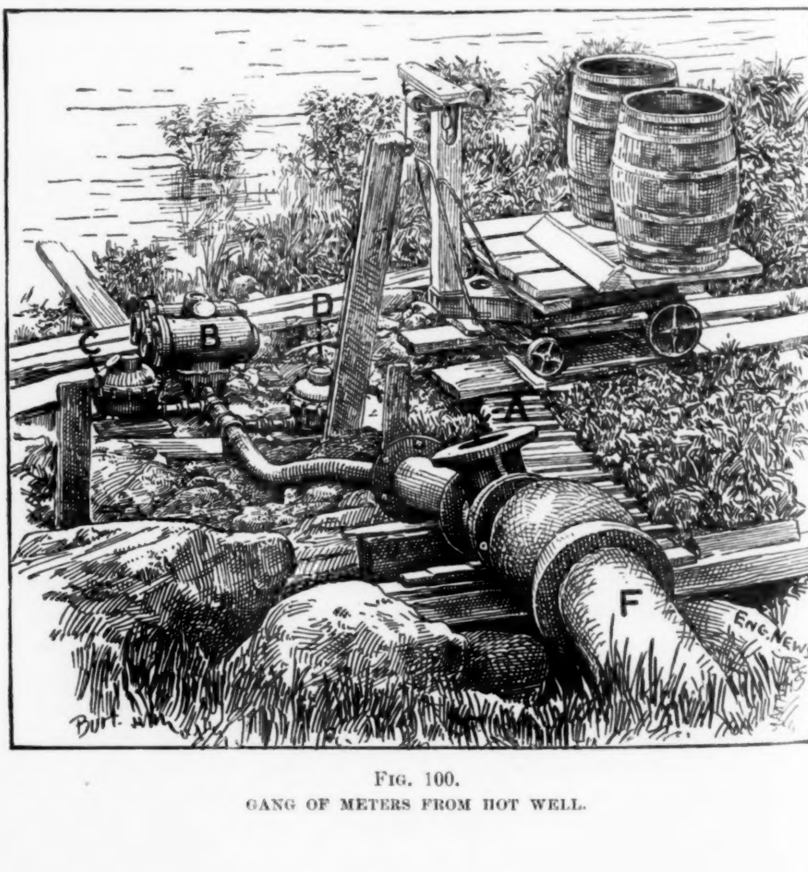

Fig. 100.
GANG OF METERS FROM HOT WELL.

in the middle is a 3-inch Worthington meter. *F* is the end of the hot well discharge pipe, as ordinarily used, and from thence to the meters is the temporary extension for the purpose of the test. *A* is an air vent to avoid the passage of air through the meters. The temperature of the water when it reached the meters did not exceed 90 degrees F., so that no trouble was experienced in operating the Crown meters. The flow of water was very uniform throughout the test. A slight amount of vapor escaped at *A*. As these meters were only under the pressure due to the eight feet of fall from the point *F* to the river, they could be tested in the ordinary way. This was done at the testing room of the Pawtucket Water Works, and the result showed that the Worthington meter registered one cubic foot for each 62 lbs. at 90 degrees F., and each of the Crown meters one cubic foot for each 63.4 lbs.

DISPOSITION OF THERMOMETERS AND GENERAL ARRANGEMENT OF APPARATUS.

In Fig. 101 is shown a sectional view of the engine, with the various thermometers and meters attached. Thermometer *G* and the meter valves, etc., are the same as in Fig. 99. Thermometer *M* was a special one, 36 inches long, graduated by H. J. Green. Twenty inches of length were devoted to the 100 degrees between 300 and 400 F. Each degree was therefore represented by a fifth of an inch, thus making the temperature of the steam measurable with great accuracy. This thermometer remained at about 356.1 degrees during most of the experiments, thereby showing the steam to be superheated about 3 degrees.

For a short time the chimney temperature rose from 428 degrees, its normal amount, to 500 degrees, and the superheating then became equal to 18 degrees, but the average was 3 degrees, and the minimum 2½ degrees.

The quality of the steam was therefore determined to be perfectly dry on entering the engine, without the application of the more or less vexatious devices known as calorimeters, for the use of one of which, the Barrus superheating form, arrangements had been made.

The thermometer *J* determined the temperature of the water from the suction well on its way into the condenser.

Thermometers *H* and *I* measured the temperature of the water leaving the condenser. *H* extended just through the pipe and *I*

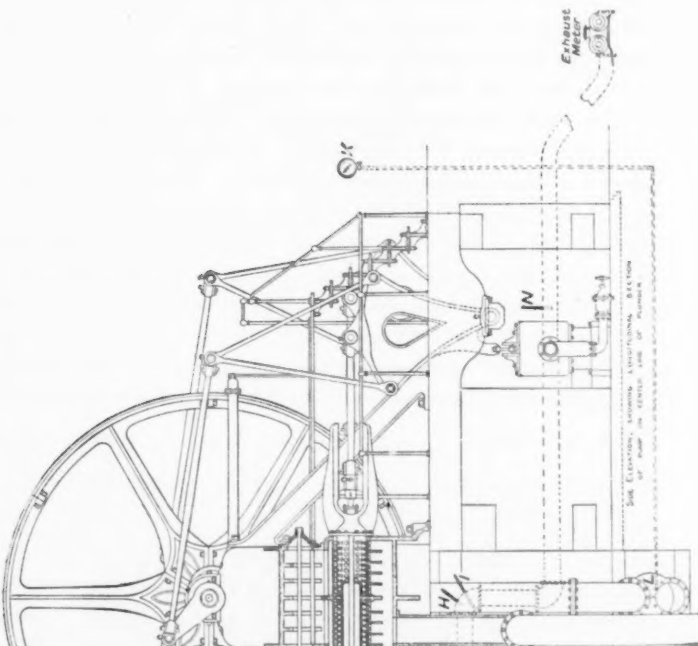

Fig. 101.
SECTIONAL VIEW OF ENGINE.

to nearly the opposite side of the pipe; the former showed constantly about 104 degrees when the latter read about 92 degrees. This apparent anomaly was finally explained by the conclusion

that the thermometer *H* showed the temperature of saturation of the steam or vapor at the pressure given by the vacuum gauge $28\frac{1}{2}$ inches, while *I* gave the temperature of the mixture of condensing water and condensed steam.

Thermometer *N* showed the temperature of the hot well water. Its readings were practically identical with those of *I*, and practically with those of *G* when the latter were not influenced by the entrance of the condensed steam from the jackets into the feed-pipe. Such steam was pumped into the feed-pipe at a point just below *M*.

MANNER OF COMMENCING AND ENDING THE FIRST TWO TESTS.

The engine being at work under its regular conditions of about 127 lbs. boiler pressure, the steam pressure was allowed to fall by deadening the fires until it reached about 100 lbs. The engine was then stopped and the fires drawn, the grates and also the ash pits being cleaned as quickly as possible. A new fire was then laid with a weighed quantity of wood, and as soon as the steam pressure recorded 130 lbs. the engine was started, and the test was under way. The water-level in the boilers was noted at the time of lighting the fires. At the end of the time allotted to the test steam is brought to as nearly as possible the pressure at which the wood fire was lighted, and the water-level is also made the same. Care was also taken that the fire was well burned out and the pyrometer in chimney falling. At the moment set for the end of the test the engine was stopped and the fire drawn and weighed.

To illustrate how little time is lost in starting a test in the above manner, the following abstract from the log is given :

Wednesday, May 30, 1889. Commencement of 72 Hours' Test.

Time.	Remarks.	Height of water with reference to a mark on glass 4 inches above bottom of the latter.			
		Boiler Pressure.	No. 1 Boiler.	No. 2 Boiler.	No. 3 Boiler.
8.30 A. M.	Ceased firing	125	0	½″	2″
8.40 "	Drew fires	110	0	0	0
8.45 "	Engine stopped	105	0	0	0
9.22 "	Ash pits and grates cleaned, new fires lighted, 263 lbs. wood	102	1½″	¾″	1¼″
9.29 "	Steam rising	112	1″	¾″	1⅜″
9.30 "	" "	120	¾″	¾″	⅞″
9.32 "	" "	130	½″	½″	½″
9.34 "	Engine started				
	Vac. Rec.				
9.35 "	Under full headway, 29 5 lbs.		Pyrometer, 410°.		
9.40 "	" " " " 5½ "	128			
9.45 "	" " " " 7 "	127			
9.55 "	" " " " 8 "	125			
10.00 "	" " " " 8 "	128			

One thousand three hundred and fifty pounds of small coal were in the furnace at this time. The throwing on of the full load within one minute is accurately the fact. This was done several times in my presence during the above tests. The load on the engine was also reduced from 110 to 30 lbs. of water pressure, with throttle wide open, in 15 seconds. The only result was the increase of speed from 49½ to 52 revolutions. The governor has, therefore, complete control over the bursting of a main.

The use of a new fire in making a test of the performance of a boiler is the only perfectly accurate method of procedure, unless the duration of the test is to exceed, say, 100 hours. To start without drawing the fires and trust to the judgment of the eye to determine an equivalent condition at the end of a short test may involve an error of 10 per cent. in the evaporative efficiency.

DIMENSIONS OF BOILERS, METHODS OF FIRING AND SELECTION OF FUEL, ASHES, CLINKERS, ETC.

Three boilers were used. The grate surface in each boiler was a circle of 4⅓ feet in diameter and 15 square feet in area. Each boiler was a cylinder 14 feet long and 4 feet in diameter,

suspended vertically in brickwork and having 48 3-inch tubes running from end to end. The total heating surface in the three boilers was 1,231 square feet in contact with water and 508 square feet of superheating surface.

The boiler flues had been swept clean about five days previous to the test. As soon as the fires were evenly ignited the fireman was limited to 200 lbs. of coal per hour, with an extra allowance of 100 lbs. for cleaning every six hours.

So nicely was the firing under control that the furnace doors were scarcely ever opened to look at the fires between the hours. As the hour approached, the allotted 200 lbs. were loaded upon a barrow and brought close to the furnace of one boiler. The damper of this boiler was then closed, the fire door opened, and one-third of the barrow-load quickly but accurately distributed evenly over the grate, and the same operation was repeated with the other two boilers. Each fire door was open less than one minute.

When the fire was cleaned the engineer assisted the fireman, one using the slice-bar and the other withdrawing the clinkers. The cleaning, including the addition of one-third of the allotted 100 lbs. of extra coal, was accomplished in about three minutes for each fire. On several occasions the whole of the 100 lbs. of cleaning coal was not required.

The above refers to the 72 hours' test, during which the fuel was mainly stove anthracite. It is the practice of the station to screen this coal on a 1-inch square mesh, thereby freeing it of all small pieces of the size equivalent to nut coal. The small size, or that which falls through the screen, is used for banking the fires, while the larger or stove size is used for the bulk of the work.

The small screenings were used for starting the fires in the 72 hours' test up to about six o'clock, as indicated in the abstract of the log.

The refuse of the anthracite coal collected in the ash pits was 752 lbs. This was screened through a $\frac{3}{8}$-inch mesh and 110 lbs. of unburned coal obtained. This coal was put on the fires during the last or 72d hour of the test, after the firing with fresh coal had ceased. The clinkers removed during the cleaning of fires amounted to 653 lbs.

The contents of the grates at the end of the test amounted to 944 lbs.

The anthracite coal was therefore as follows :

Total coal fired	15,710 lbs.	
Wood, taken as equivalent to 40 per cent. of coal	105	
Total fuel used		15,815 lbs.
Consumption per hour	219.5	
Incombustible from ash pits	641	
Incombustible clinkers	653	
Left in furnaces	944	
Total		2,238
Total combustible		13,577 lbs.

The total refuse equals 14.3 per cent. of fuel.

The bituminous coal made no clinkers, and the fires were not cleaned during the entire 24 hours of its use. No separation of the ashes was made, the total refuse from ash pits and grates combined being 510 lbs.

The total consumption was :

Coal	5,200 lbs.
Wood	80
Total in the 24 hours	5,280 lbs.

Consumption per hour, 218.6.
Ashes equal 9.6 per cent. of fuel.

I regard the above details of firing, etc., as worthy of note, from the fact that it is a matter of well-founded opinion that a very sensible part of the performance of fuel depends upon the degree of skill displayed in the firing when the best results are obtained. Beyond doubt the extraordinary annual performance of the Pawtucket pumping engine is partly due to the high state of efficiency to which the routine firing and other details of management have been brought by Chief Engineer Walker's zeal and intelligence.

PRINCIPAL RESULTS—ANTHRACITE TEST.

Date, May 30 to June 1, 1889.
Duration, 72 hours.

Diameter of high-pressure cylinder	15	inches.
Diameter of low-pressure cylinder	30½	"
Length of stroke, both cylinders	30	"
Length of cut-off, high-pressure cylinder	0.22	"
Length of cut-off, low-pressure cylinder	0.335	"
Clearance, high-pressure cylinder	4	feet.
Clearance, low-pressure cylinder	3.7	"

Port area in per cent. of piston area :—	
Admission high-pressure cylinder	9.4 per ct.
Exhaust high-pressure cylinder	14.8 "
Admission low-pressure cylinder	5.4 "
Exhaust low-pressure cylinder	9.8 "
Diameter of all pump plungers	10.52 ins.
Diameter of all rods	$2\frac{3}{4}$ "
Revolutions per minute	49.03

Jackets envelop both cylinders, including heads. Volume of receiver approximately equal to volume of low-pressure cylinder.

BOILERS.

Number in use	3
Water heating surface	1.231 sq. ft.
Superheating surface	508 "
Grate surface	45 "
Ratio of heating to grate surface	38.6 "

PRESSURES.

Average boiler pressure	127 lbs.
Average receiver pressure	$8\frac{1}{2}$ "
Average vacuum	$28\frac{1}{2}$ ins.
Average back-pressure in low-pressure cylinder	1.25 lbs.
Average water pressure against pumps	115 "
Average suction against pumps	5.78
Average barometer pressure	30

TEMPERATURES.

Temperature of saturated steam at average boiler pressure	353 deg. F.
Average temperature of steam in pipe two feet from steam chest	356 "
Degrees of superheat	3 "
Temperature of feed-water entering boilers	104 "
Temperature of feed-water leaving hot well	92.1 "
Increase of temperature due to admission of jacket water	11.9 "
Temperature of jacket water entering feed-pipe	310 "
Temperature of condensing water leaving suction well	63 "
Temperature of gases at exit from boiler flues	428 "
Temperature of gases after passing heater in flue through which condensation from receiver was circulated	354 "
Temperature of water blown to waste from receiver	225 "
Temperature of the atmosphere	70 "

TOTAL QUANTITIES.

Anthracite coal	15,710 lbs.
Equivalent of wood at 40 per cent., 263 lbs	105 "
Total	15,815 "

Ashes, clinkers, etc.	2,238 lbs.
Water fed to boilers, including condensation from jackets	140,969 "
Condensing water	4,201,337 "

HOURLY QUANTITIES.

Fuel	219.5 lbs.
Feed-water	1,958 "
Condensing water	58,351 "
Condensed in jackets	107 "
Condensed through receiver	156 "
Blown to waste from receiver	118 "
Average indicated horse-power, steam cylinders	143.89
Average indicated horse-power of pumps	138.36
Average horse-power to overcome friction of engine, including air pump	5.13
Average horse-power to operate air pump	2.35
Rate of combustion or coal per square foot of grate per hour	4.9 lbs.

ECONOMY OF ENGINE.

Steam per hour per indicated horse-power of steam cylinders	1,364

This figure should be regarded subject to a possible error of 3 per cent., due to this limit of error affecting the determination of horse-power from indicator cards.

Efficiency of mechanism of engine pumps	96.4 per ct.

ECONOMY OF BOILERS.

Water evaporated per pound of fuel from 102 degrees into steam at 125 lbs. pressure	8.88 lbs.
Equivalent evaporation from and at 212 degrees per pound of combustible	12.12 "

ECONOMY OF ENGINE AND BOILERS COMBINED.

Duty per 100 lbs. of coal at actual efficiency of boilers	124,720,000 ft. lbs.
Coal per hour per indicated horse-power of steam cylinders	1.54 lbs.
If boilers had evaporated 10 lbs. of water into steam from actual temperature of feed-water, the duty would be per 100 lbs. of coal	140,430,000 ft. lbs.
Coal per hour per indicated horse-power of steam cylinders	1.37 lbs.

CHECK UPON MEASUREMENT OF FEED-WATER BY MEASUREMENT OF HEAT EXHAUSTED OR REJECTED AT AIR PUMP OF ENGINE.

The heat which entered the engine per hour was as follows: 1,958 lbs. of feed-water, to each pound of which there was supplied 1,221 heat units, the total heat of evaporation, hence:

Heat given to steam, 1,958 × 1,221 =	2,390,718 heat units.	
The chimney heater evaporates 38 lbs. of water at 225 degrees in steam at 22 lbs. pressure, making 38 × 960 =	36,480	
Total heat received by engine.........		2,427,198

This heat is distributed as follows:

58,351 lbs. of condensing water is raised in temperature from 63 degs. to 92.1 degs. F., making........	1,698,014	
118 lbs. of the feed-water is blown to waste from the receiver, representing 118 × 225 =	26,530	
107 lbs. of the feed-water is drained from the jackets, representing 107 × 310 =........	33,520	
The remainder of the feed-water, or 1,958−118−107 = 1,733 lbs., is found in the hot well, representing 1,733 × 92.1 = ...	159,609	
The indicated work performed, or 143.49 horse-power represents $\frac{143.49 \times 1{,}980{,}000}{772} =$	368,769	
The estimated radiation is 20 lbs. of steam per hour..................................	38,320	
		2,324,412
Balance unaccounted for		102,786

This discrepancy represents 4 per cent. of the whole amount of heat. Part of this is ascribable to the leakage of steam from the safety-valves and stuffing boxes referred to above, and the vapor which escaped from the vent *A*, Fig. 100. The remainder is due to undefined causes. The error is, however, on the right side to make the feed-water measurements worthy of confidence as being slightly too large and unfavorable to an exaggeration of the economy of the engine.

BITUMINOUS TEST.

All data for this test are the same except the following:

Revolutions per minute..............................	49.44
Average boiler pressure..............................	128 lbs.

Average water pressure against pumps	116 lbs.
Average cut-off high-pressure cylinders	0.235
Average horse-power indicated, steam cylinders	147.95
Average horse-power indicated, pumps	140.74
Average horse-power indicated, devoted to friction	7.21
Total bituminous coal	5200 lbs.
Total wood = 40 per cent. × 200 =	80 "
Total	5280 "
Total ashes	510 "
Duration of test, 24 hours 13 minutes	
Fuel per hour	218.6 "
Feed-water per hour	2046 "

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Steam per hour per indicated horse-power of steam cylinders	13.82 lbs.
Efficiency of mechanism	95 per ct.

ECONOMY OF BOILERS.

Water evaporated per pound of fuel from 102 degrees into steam at 126 pounds	9.35 lbs.
Water evaporated per pound of combustible, and at 212 degrees	12.11 "

ECONOMY OF ENGINE AND BOILERS COMBINED.

Duty per 100 pounds of coal at actual efficiency of boiler	127,350,000 ft. lbs.
Coal per hour per indicated horse-power of steam cylinders	1.48 lbs.

ECONOMY OF ENGINE AND BOILERS COMBINED IF BOILERS HAD EVAPORATED 10 POUNDS OF WATER PER POUND OF COAL FROM TEMPERATURE OF 102°.

The duty per 100 lbs. of coal =	145,041,000 ft. lbs.
Coal per hour per indicated horse-power of cylinders	1.30 lbs.

By reference to the remarks on the theoretical heating capacity of the two kinds of coal it will be seen that the increase of duty with the bituminous coal was not as great as the chemical analysis indicates should have been the case.

INFLUENCE OF STEAM-JACKETS.

The engine was worked for six hours, and again for nine hours, without the presence of steam in the jackets, but with the flue

reheater in action. The coal consumption was not determined, the steam consumed, as shown by the feed meter, being the desired measurement. For six hours the total water consumed was about 180 cubic feet. The level of water in the boilers could be made the same at the beginning and end of the test, within one-half inch. Thereby the maximum error of water determination would be, for three boilers, about one-half of one per cent. No practical error is involved, therefore, in basing conclusions upon so short an interval as six hours. Table IV. shows the principal results compared with those obtained with the jackets in use. Column 14 gives the steam consumed per hour per horse-power. There is apparently a gain in economy by the use of the jackets, the consumption being 13.62 lbs. with their use, and as high as 14.17 lbs. without them.

But the differences in cut-off in the high-pressure cylinder represent differences of consumption, which, theoretically, may cause the variations in economy. Furthermore, the probable error in mean pressure determinations from the indicator cards may be two per cent. It is not possible, therefore, to conclude that the differences of economy shown in column 14 are entirely attributable to the influence of the jacket. As a probable result, the average of the two tests, with and without the jacket, may be compared. This basis makes the gain in economy due to the jacket equal to about three per cent. Inasmuch as the complete drainage of the jackets is regarded as important in temporarily dispensing with the use of the latter for the purposes of a test, the following explanation is made regarding the steps taken to insure the emptiness of the jacket spaces when the latter were not supplied with steam. The pipe supplying the steam to the jacket is one inch in diameter. It connects with the jackets at the bottom of the cylinders, and there are no obstructions to prevent condensed water from flowing by gravity from the jackets into the well of the jacket pump, situated about one foot below the bottom of the cylinders. When the jacket is supplied by steam, a "straight-way" valve on the one-inch pipe is wide open to admit the steam from the boiler.

When the test without jackets was made, this valve was closed, and then a one-inch valve on the same pipe, connecting with the waste pipe, was opened wide, thereby causing the jacket spaces to be violently blown free of all water then in them. The connection between the jacket pump and the boiler feed pump was

then closed by a globe valve, and the delivery side of the pump opened to the atmosphere. The pump remained in action, and the blow-off valve remained open during the tests. The latter were not commenced until the engine had been running about an hour under the above conditions.

Table V. compares the theoretical steam consumption of two of the four tests. Columns 10 and 11 show that eighty-eight per cent. of steam at release in high-pressure cylinder appears at cut-off in low-pressure cylinder with the jackets, and eighty-four per cent. without jackets. The re-evaporation is about eight per cent. of the total consumption in both cylinders with the jacket, and about half as much without the jackets. The difference of re-evaporative effect with the jacket in use represents just about the steam condensed in the jackets. This fact can hardly be regarded as proven, however, so far as the indicator cards are concerned, for the latter do not show any uniform difference in the law of the expansion line with and without jackets, respectively. This is evident from Fig. 51, in which the dotted points between the lines $S\ S$, $S_1\ S_2$, etc., represent the expansive lines of different indicator cards reduced to the same cut-off and length, all high cards passing through A and all low cards through P.

The indicator cards were taken with all ordinary care and skill, and with a truly parallel motion—a pantagraph or Corliss lazy tongs being used. But no precautions were taken to eliminate the finer elements of error in the drum* motion. Consequently the length of cards differed over a range of 2 per cent. in about five inches. Such error is not important for most other deductions, but it is inadmissible where such a refinement as the influence of jackets is to be studied. Arrangements are now being made to obtain cards with and without jackets in use, by the use of more perfect indicator drum motion.

CONDITION OF ENGINE AND BOILERS.

The engine was placed successively on the several dead centres and the leakage of the inlet valves and pistons noted by observing the escape of steam from the proper indicator cocks. The greatest leakage was measured by attaching a hose and leading the steam into a condenser. From this experiment the leakage in the other cases was estimated by the eye.

* See discussion of such errors in paper by J. B. Webb. Trans. Am. Soc. M. E.

LEAK THROUGH INLET VALVE.

Low cylinder head,	E............6 lbs.	per hour, by experiment.
Low cylinder crank,	E............2 "	" " estimate.
High cylinder head,	E............2 "	" " "
High cylinder crank,	E............1 "	" " "

LEAK THROUGH PISTON RINGS BY ESTIMATE.

Low-pressure cylinder head, E..........................	2 lbs.
Low-pressure cylinder crank, E..........................	4 "
High-pressure cylinder head, E..........................	6 "
High-pressure cylinder crank, E..........................	0 "

The exhaust valves could not be conveniently tested. Two of the safety-valves leaked slightly, but it was not considered judicious to stir them. A leak was present at the high cylinder piston rod, and at the plunger of the jacket pump. No allowance was made for these leaks in the steam consumption.

COMPARISON OF PRESSURE MEASUREMENTS AND TEST OF PRESSURE GAUGES AND INDICATOR SPRINGS.

The Ashcroft Manufacturing Company kindly loaned me for this work six of the latest designs of the Tabor Indicators, which were sent directly from their works to the pumping station. Four of these indicators were fitted with springs standardized for use under cold water pressure, and the remaining two, together with a pair from the Winters cabinet, contained springs standardized for steam contact. The indicators with water springs were tested on an Ashcroft gauge tester; the latter belonged to the pumping station, and consisted of a hollow tube filled with glycerine, to one end of which the instrument to be tested could be attached, while at the other end a known pressure can be produced by loading a piston with weights. The piston has an area of 0.248 square inches, and weights were provided corresponding to each five pounds pressure per square inch. I found these weights in accurate condition, and that the water springs of the indicators agreed with the testing apparatus, and with each other, and with the pressure gauges from which were read the water pressure against the pumps—within one pound per square inch, which was the limit of accuracy of the testing apparatus.

The gauges in the engine room from which the boiler press

ures were read were also tested on the Ashcroft tester, with the following results :

Reading of Gauges.	Reading of Standard.
$32\frac{1}{2}$ lbs. per sq. in.	30 lbs. per sq. in.
57 " "	55 " "
82 " "	80 " "
$107\frac{1}{2}$ " "	105 " "
123 " "	120 " "
128 " "	125 " "

Now the gauge is attached to the vertical portion of the steam-pipe in such a position that there is a column of condensed steam, 4.62 feet in height, which interposes its weight between the mechanism of the gauge and the boiler pressure, the gauge therefore fails to receive the full steam pressure by the weight of the 4.62 feet of water, which amounts to 1.93 lbs. per square inch. Calling this two pounds and applying it as correction to the above figures, it appears that :

When the steam reads	$30\frac{1}{2}$	the boiler pressure is	30 lbs.		
" "	55	" "	55 "		
" "	80	" "	80 "		
" "	$105\frac{1}{2}$	" "	105 "		
" "	121	" "	120 "		
" "	126	" "	125 "		

The discrepancies are all within the limit of error of the testing apparatus. Mr. Walker, the chief engineer, had informed me that he had adjusted the gauge to allow for the effect of the water column, and I give the above detailed account as an excellent illustration of the skilful accuracy with which the details of the pumping stations are maintained.

The steam springs of the indicators were tested by using the different indicators and springs at the same cylinder cock at times when the conditions of boiler, receiver and vaccuum pressures were the same ; the fact that they all agreed with each other, and that six of them were directly from the hands of a standard manufacturing company, was considered sufficient evidence of accuracy.

The vacuum gauge was not tested ; the average vacuum was $28\frac{1}{2}$ inches, the average barometer reading was 30 inches, and

the average back-pressure in the low-pressure cylinder $1\frac{1}{4}$ pounds per square inch, or $2\frac{1}{2}$ inches of mercury. These figures make the back-pressure due to imperfect vacuum equal to $30 - 28.5 = 1.5$ inches $= 0.76$ pounds per square inch, and due to forcing steam out of cylinder 0.49 pounds per square inch. There is, therefore, no occasion to regard the vacuum gauge as possessing any error of practical importance.

The suction pressure against the pumps was shown by a float, which gave the feet of height through which the water was lifted from the well. This height averaged 13.35 feet; the temperature of the water lifted averaged 63.6 degrees, F.; hence the suction pressure was 5.78 pounds per square inch. The suction pressure was also measured by a vacuum gauge connected to the pump chamber at the centre of the latter height; the average suction pressure by this gauge was 5.65 pounds per square inch.

In the 72 hours' test the average pressure shown by the water pressure gauge K (Fig. 3), connected at the discharge pipe at a point between the pump chamber and the main check valve, was 109.69 pounds per square inch, reduced to the level of the centre of pump pistons. Adding this to the average of the two figures for suction pressure given above, we have the total head pumped against, as determined by pressure gauges, equal to 115.4 pounds per square inch. This pressure should be less than the mean effective pressure shown by the pump indicator cards, by the amount equal to the force required to work the pump valves, but the mean effective pressure shown by the pump indicator cards was from 114.7 to 115 pounds per square inch. This paradoxical discrepancy was apparent early in the test, and was made the subject of the most careful study. But while the errors involved in calculation of gauges and springs, and the fact that the very fine scale of 100 had to be used for the pump cards, may allow the discrepancy between the two figures to be set aside, there is still no surplus to represent any sensible loss of pressure due to working of the valves. In the calculations of horse-power and duty I have used the mean of the pressures deduced from the gauges and indicators respectively.

THEORETICAL HEATING CAPACITY OF COALS USED IN TESTS.

Chemical analyses of the anthracite coal used in the 72 hours' test, and of the bituminous used in the 24 hours' test resulted as follows:

	Anthracite.	Bituminous.
Moisture	1.80 per cent.	1.03 per cent.
Hydrogen	4.60 "	5.85 "
Carbon	79.30 "	83.00 "
Sulphur	0.85 "	0.72 "
Oxygen	4.65 "	1.50 "
Nitrogen	1.90 "	1.45 "
Ash	6.90 "	6.45 "
	100.00 "	100.00 "
Ashes, clinkers, etc., by test	14.3 "	10.00 "

Assuming that the excess of ash in the boiler furnace over that given by chemical analysis is unburnt carbon, we have the heat evolved by combustion as follows, in British thermal units:

	Anthracite.	Bituminous.
Hydrogen	2,512.00	3,660.00
Carbon	10,425.00	11,534.00
Sulphur	33.00	32.00
	12,970.00	15,226.00

Besides the moisture shown above, it is probable that each coal contained 3 per cent. more of moisture during the test, which dried out of the coal during the interval of a month intervening between the time of the test and that of the analysis.

A sample of the anthracite coal dried over the boiler for five hours lost 3 per cent., but no similar determination was made in the case of the bituminous.

The heat evolved by combustion is distributed as follows:

1. Each pound of moisture requires 1100 heat units to evaporate it from 78°, the temperature of the boiler room, to steam at atmospheric pressure, and also 108 units to superheat the vapor to 428°, the temperature of the chimney. There would thus be consumed 58 thermal units for the anthracite, and 47 thermal units for the bituminous coal.

2. Each pound of anthracite coal evaporated 8.88 pounds of water from 103° temperature of feed into steam at 127 pounds gauge pressure, and similarly the bituminous evaporated 9.35 pounds of water. There would thus be consumed 9919 thermal units for the anthracite, and 10,444 thermal units for the bituminous.

3. Chemical analyses of the chimney gases in the case of the anthracite gave the following result:

PER CENT. BY VOLUME.

No.	Carbonic Acid.	Carbonic Oxide.	Oxygen.	Nitrogen by Diff.
1.	8.2	0.4	9.4	82.0
2.	8.5	0.4	12.1	79.0
3.	9.5	0.0	10.9	79.6
Average,	8.7	0.3	10.8	80 3

From these figures it follows that with the anthracite the loss of heating effect due to imperfect combustion by formation of carbonic oxide was 132 thermal units per pound of coal.

4. The above analyses of chimney gases give for the average weight of air supplied per pound of carbon 25 pounds, which is roundly equivalent to 20 pounds per pound of coal. Adding 0.85, the combustible in a pound of coal, we have the weight of gases escaping by the chimney as 20.85 pounds per pound of coal.

By measurement of the velocity of air entering the ash pit doors, with a correct Cassella anemometer, it was determined that the average velocity for all parts of the area of opening corresponded to the supply of 20.28 pounds of air per pound of coal, to which the addition of the combustible being made, we have 21.13* as the pounds of chimney gas per pound of coal.

Taking 21 pounds as the mean of the two methods of determination, we have the heat escaping by the chimney at 428° temperature—1,954 thermal units per pound of coal.

Summarizing these expenditures we find the total for the anthracite is 12,263 units, leaving 12,970 − 12,063 = 907 units as the effect of radiation from the external surface of the boilers and from ash pit.

Similarly the total for the bituminous is 12,577, leaving 15,226 − 12,577 = 2649 units for radiation.

As the coal burned per hour was 220 pounds for both kinds of fuel, it is not reasonable to suppose that over twice as much radiation could occur with the bituminous as with the anthracite. Allowing 1½ heat unit per hour per square foot of surface per degree difference of temperature as the radiation co-efficient, and taking the average difference of temperature between the boiler

* The brick setting of the Corliss b ilers was almost entirely encased in iron, and the brickwork was in excellent condition. The openings for air supply above the grate through the fire door amount to but one per cent. of the ash pit openings, and are therefore neglectable in the presence of other sources of error.

surface and the atmosphere as 200°, the possible loss by radiation would be 150,000 thermal units per hour, the boiler surface being about 750 square feet. This amounts to 1050 units per pound of fuel. It is therefore probable that the radiation loss for the anthracite is nearest correct, and that therefore the bituminous coal contained more moisture, gave more carbonic oxide than was found for the anthracite, or that the air per pound of coal may have been more, making the chimney loss greater.

The probable distribution of heat may therefore be summarized as follows:

	Anthracite.		Bituminous.	
	Heat Units.	Per Cent.	Heat Units.	Per Cent.
Consumed in making steam..........	9919	76.5	10,441	68.6
Lost in evaporating moisture.........	58	0.5		
Lost by imperfect combustion........	132	1.0	3,878	25.4
Lost by chimney....................	1954	15.0		
Lost by radiation......	907	7.0	907	6.0
Total by chemical analysis........	12,970	100.00	15,226	100.00

NOTE.—For fuller details see *Engineering News*, commencing December 28, 1889, page 603.

DISCUSSION.

Mr. Chas. E. Emery.—The impression left upon the mind after reading this paper is that the saving due to steam-jacketing in general is but 3 per cent. at the greatest, whereas it has been shown distinctly in other cases that there is a saving of about 12½ per cent. The result stated in the paper must therefore be looked upon as a special case. These engines are quite familiar, and were undoubtedly used with the Corliss boiler, with superheating surface formed by vertical tubes extending above the water. A very slight superheating would account for the absence of gain by a steam-jacket in the first cylinder, and the presence of a reheater in the intermediate chamber for the absence of gain in the second cylinder. The results due to superheating, those due to reheating, and those due to the jackets are all of the same kind, and in the same direction. All these processes simply reduce the amount of cylinder condensation by keeping the internal surfaces of the cylinder nearer to the initial temperature of the steam. It is true, as the author says, that the steam-jacket is an imperfect means of transferring heat. This should be considered as a

particular case for the reason that where there is no superheating and no reheating, a very appreciable gain has been shown by the use of the steam-jacket.

Mr. C. A. Hague.—I gather from the paper that the difference in the consumption of steam with and without the jackets has been measured from the indicator cards. Now it seems to me that, with a pumping engine driving such a constant load so easily measured, it would give more accurate results to base the comparison upon the actual work done. The errors of indicators, and the variation of moisture in the steam resulting from superheating or over-saturation, bring factors into the question which render it very difficult to arrive at a conclusion in the matter; and I would like to ask the author how he determined these quantities of water or steam, the weight of steam per horse-power, and how he determined the steam used by the engine, with and without the jacket. An engine pumping water, or an engine driving a flour-mill, is under such a constant load that it seems as though the work was very easy to measure outside of the indicator itself. The water accounted for by the indicator in a good class of unjacketed compound engines at high piston speed does not go much above 78 per cent., and with triple expansion and the higher class of compound engines the efficiency or the water accounted for drops below that.

I had an experience two or three years ago in testing a pumping engine on reservoir service almost identical with this. The total load was 100 lbs. to the square inch. On one of the engines the jacket trap got out of order in the middle of the test. The engine then was indicating 107,000,000 duty by actual weight of water without deduction, and the jacket trap getting out of order we shut it off, and the duty fell to 99,000,000. Well, now, of course there might have been some water left in the jackets, which would lower the economy when the jackets were not in use, but I have reason to believe they were empty. We shut off the jacket pipe and took advantage of the opportunity for a few minutes to see what the difference would be in the duty; and gradually as we observed it it fell from 107 to 99, which is a difference of over 8 per cent.; and, as already stated, before forming any opinion I would like to know how the quantity of steam used is determined, and whether powers are determined by the indicator, or whether they are determined by the water end of the machine.

Mr. Oberlin Smith.—As I understand this jacket matter, the difference between the results with and without a jacket are when respectively the steam is allowed to circulate through the jacket and when it is merely shut off by cocks or valves, leaving the jacket "empty"—that is, full of air or vapor at a very low pressure, or whatever else may be in there. I do not know much about the subject experimentally, but it seems to me that this is hardly a fair test. In order to show the absolute difference of effect between jacket and no jacket, the jacket should be used in the ordinary way in the one case, and in the other case should be entirely removed. Otherwise its contents act to some extent as a non-conducting "lagging." I would like to ask the author whether he has tried any experiments in this way. It would be an easy matter to make a jacket in halves, or to make it so that it could be slid off endwise, or something of the kind, so that the experiment could be tried with all other conditions the same. It would not be a fair trial to take two separate engines, built as nearly alike as possible except in the matter of jacket, because the conditions of friction, etc., might be different. But with a removable jacket the thing could be very accurately tried and the real results arrived at.

Mr. A. R. Wolff.—The idea of the steam-jacket, as it has always been understood, is not to prevent the radiation of heat from the cylinder, but to supply heat units to the steam during its expansion in the cylinder, so that the rate of condensation there will be decreased. It is well known that with a certain amount of water in the cylinder during admission the amount will increase very rapidly during expansion. The idea of the steam-jacket is to supply steam to the jacket and have the condensation take place within the jacket itself, and thus transfer heat to the steam in the cylinder, during its expansion, rather than to have the condensation take place in the cylinder itself. On that account the trial is certainly entirely fair for the determination of the relative efficiency of the engine with and without a jacket. The presence of the jacket when empty and not used, incapable of radiating heat to the steam in the cylinder (since it contained no available heat units), would not cause the engine to show a higher efficiency than if no jacket was round the cylinder. The point of this paper, as I understand it, is the following: that with and without jackets in this particular engine the steam [illegible]tion certainly did not differ

in excess of 3 per cent.; that furthermore, the variation in the indicator cards taken approximately under the same conditions is such that it makes it doubtful whether this variation in the performance is not due to variation in the indicator cards. Now, if that be so, the possibility of such variation may enter all the steam-engine trials thus far on record to show the relative value of jackets, because nearly all—if not all—the leading experiments in steam-jackets were made on steam-engines in which the indicated horse-power and the steam consumption formed the record to test the efficiency. These percentages of difference, to which reference has been made in discussion, viz., 11 and 12 per cent. gain by the use of the jacket, were found on the trials of engines in which the indicated power was measured and the actual steam consumption obtained, either by the evaporation of the boiler and testing the quality of the steam, or by measuring the heat units in the steam as it left the cylinder. Certainly the author's experiments now communicated to the Society were made with the greatest care, and whatever bearing the results may have on other experiments, it seems unfair to question the correctness of the special results on this particular engine, because of some other experimental records with other engines not here detailed.

I may add that I have carefully examined, especially with a view to this discussion, other records above referred to, in which 11 and 12 per cent. gain have been recorded as obtained, and as far as I have been able to examine those records, the indicated powers differed so widely within a given set of conditions with jackets, or again without, that it made these records of 11 and 12 per cent. gain and the like appear very dubious, indeed, as to accuracy and justification.

Mr. J. T. Henthorn.—I have in mind a series of experiments which were made with an engine of about the same type as the "Pawtucket," running at the same ratio of expansion, with the same steam and water pressure and jacketed in the same way, and it was found from a series of twenty-four and thirty-six hour runs, repeated alternately, that the value of the steam jackets was from 7 to 10 per cent. That is, that was the value of the jackets to the economy of the engine and based on the actual duty of the engine in foot pounds.

Prof. J. B. Webb.—In some of these tests of the efficiency of jackets it seems that doubts exist as to the accuracy of the

indicator cards and other data, and it seems to me that any supposed saving due to the jacket must remain doubtful until it be made reasonably clear from the theory why the jacket should save, or until such saving be demonstrated by an unimpeachable set of experiments. It is a question whether a certain amount of steam can be used more economically in the jacket than it can in the cylinder. One of the first conditions in using steam to get power out of it is that no fall of temperature shall be allowed without a certain proportion of power being developed. The steam in the jacket of the low-pressure cylinder must be subjected to a considerable fall from the boiler temperature to that of the cylinder, so that there is at once a great loss for the steam which is used in that jacket. There is a way in which it may be seen that the jacket might save. The condensation of the steam in the cylinder, as has been suggested, I believe, by Rankine, probably obeys the same law as the freezing and boiling of water—viz., that the temperature at which it occurs changes with circumstances, and it may be that a jacket acts mainly to delay the formation of the first particles of condensed water. As long as their formation can be prevented condensation cannot occur, but as soon as they form they act as nuclei and it goes on rapidly. The action of a jacket may resemble stopping a leak in a dam, which, left unstopped, enlarges its hole, and may finally cause a loss out of all proportion to the small beginning. When condensation has once commenced, it is doubtful whether any reason can be assigned why the steam can be used to more advantage in the jacket than in the cylinder itself, for the reason that a loss of temperature occurs equal to the difference between boiler temperature and the average temperature of the inside of the cylinder. In view of this unavoidable loss, it seems to me doubtful whether it can be shown that the jacket effects a saving.

Prof. R. H. Thurston.—These results are interesting—in some respects extraordinary; they surprise many of us.

The result which has been shown in the author's paper is one which has not been entirely unanticipated as an abstract possibility. Our experience with jackets runs back now a great many years. With the old single cylinder engine Watt found—and I think a good many engineers since his time have found—that by the use of steam-jackets, under the usual conditions, it was perfectly possible to make a difference of 15 or 20 per

cent. in the duty obtained from such an engine. In the use of compound engines, more recent experiments have shown that a difference of 10, or perhaps more, per cent. may be produced by the proper use of jackets. The statement—which is undoubtedly true—that an engine of such extraordinary excellence of construction, such remarkably efficient design, as to give a record which, at the time of its construction, had never been approached, could still give equal efficiency without the use of a jacket is to me one of the most interesting and instructive facts which has yet been experimentally revealed. But, noting the several facts to which I have referred in my own paper and here, it can be seen that the effectiveness of the jacket is of progressively decreasing magnitude as the perfection of our provisions for increasing efficiency in other directions increases.

In one of my own papers, I think, you will find a remark that, under certain conditions, a jacket may be of no value, and under certain other conditions may even exaggerate the losses which it was intended that it should reduce. I can imagine a condition in which, in this engine, there should be no saving, but that the use of the jacket should give us a decreased efficiency. I should be surprised if it should be so, but I can certainly imagine an engine so reducing efficiency by the use of the jacket.

. Accounts of the performance of the Chicago "West-Side" Pumping Engine report duties of ninety-five and ninety-seven millions, and that with compound engines unjacketed, except on their lower heads. Mr. Wilson, their designer, told me that he had taken the view which I have just presented to you; that he was convinced that he could build a cheaper engine, and get as good duty when jacketing simply the lower heads as when jacketing all over. He *did* build an engine which for the time gave extraordinarily high duty; and he built a cheap engine. He put on a jacket which proved to be effective, and a form of jacket which could be made without increasing the cost of the machine very much, or introducing the risks which always are introduced by the application of a jacket of the ordinary kind.

Prof. De Volson Wood.—I notice that the steam consumption of the engine appears to be remarkably low. I have watched with some interest the results of trials, and I find that by adding 3 per cent. to this that it is still lower than any which I have considered authentic. I do not say this by way of criticism. I take it for granted that the experiments are well made,

but I do it rather to call attention to the fact and also to draw out, in case I am mistaken, in regard to steam consumption from others, the fact that these figures have been approximated to. I would like to have it confirmed.

Another point: Admitting the correctness of the experiments and the results, it would seem to show in this engine that the empty steam-jackets might have been serviceable. We cannot tell definitely, I suppose, without some such experiment, as was referred to by Mr. Smith, in which the whole steam-jacket is removed. But it is possible that the space once having been heated and filled with hot air, did all the service which the live steam-jacket could have done. I was informed by the engineer of the Cromwell Line that they were running single cylinder engines of long stroke and early cut-off (about $\frac{1}{10}$), in which there was an arrangement for a steam-jacket. He said: "I have not used a steam-jacket for years."

I asked him why not. He said, "It seems to give us as good results as when we used live steam." He supposed that the space being filled with air formed a hot-air space which prevented transmission of heat outward. If this be true, it follows that making the cylinder practically a non-conductor of heat, it is in this case as beneficial as the attempt to transmit heat from the jacket to the cylinder.

Major English, who has made an extensive series of experiments upon the action of steam in cylinders, draws several conclusions, among which is the following: "Reducing the area of unjacketed clearance surface as far as possible, appears to the author to be unattended with any counterbalancing disadvantages; and he is of the opinion that it is of greater importance effectually to jacket the cylinder covers and piston than the sides of the cylinder themselves; and that the economical results obtained by the Corliss and other similar valve-gear are more directly attributable to short steam passages and consequent reduction of clearance surface than to any other cause." *

Mr. John C. Kafer.—The amount of steam used per indicated horse-power, as shown by the figures, is apparently very low, and this would imply that the steam was superheated, and if highly superheated there would be very little cylinder condensation, and the efficiency of the jacket would be proportionately decreased.

* *Engineering*, November 15th, 1889, p. 586.

Experiments were made twenty-five years ago, showing the advantage of superheated steam—if it could be practically used —the object being the same as that sought after by the use of the steam-jacket.

It is a fallacy to suppose that the use of the steam-jacket will prevent condensation; steam will be condensed in one case in the jacket, and in the other in the cylinder, the difference being that when condensed in the cylinder re-evaporation takes place on the reduction of pressure, and when condensed in the jacket, the pressure being constant, there is no re-evaporation.

If the steam is superheated, there would be very little cylinder condensation in this type of engine, and the amount of steam accounted for by the indicator would approximate closely to the quantity of steam used by the engine.

I should like to know what percentage of the steam used is accounted for by the indicator with and without the jacket in use.

The President.—Before Professor Denton sums up the discussion I would call attention to one other point, namely: that the economy of the steam-jacket is undoubtedly affected by the amount of expansion carried on within one cylinder—that is, by the difference in temperature at the two ends of the stroke; and that this might explain the fact that different results would be obtained from an engine having a very high degree of expansion in a single cylinder, and a compound engine having a moderate degree of expansion in each of several cylinders.

*Prof. Jas. E. Denton.**—I do not desire the impression created by my paper to be that the saving by jackets is limited to 3 per cent. in *all* classes of engines. I am skeptical, however, about much greater saving ever having been proved to have been caused by jackets on engines having a speed of revolution upwards of 60 per minute. Will Mr. Emery name any experiments which, in his opinion, have proved to the contrary?

Mr. Emery.—I had in mind those made on the *Gallatin.* There was the same check upon the results with that vessel as in testing a pumping engine. It is to be recollected that the experiments were made under the general direction of Chief Engineer U. S. Loring, U. S. N., representing the U. S. Navy, and myself, representing the Treasury Department. The vessel was secured to a dock in a slip, out of the influence of the tide, and a constant draft of water maintained by keeping the supply of coal on the

* Author's Closure, under the Rules.

dock, and only bringing it on board in bags as required from time to time. It follows, therefore, that the resistance was the same during both experiments, and measured practically by the number of revolutions of the propeller in each case, so that the economy of the steam-jacket can be ascertained by comparing the actual amounts of water evaporated and coal burned with the number of revolutions of the propeller, entirely independent of results obtained by comparing the power shown by the indicator diagrams.

I would only add that all deductions from scientific researches can be invalidated on this principle of stating that the change in result due to a particular modification is within the limits of error of observation, or of the instruments used. Deductions must necessarily be made from averages in experimental matters, the same as in ordinary commercial transactions.

Prof. Denton.—The experiments on the *Gallatin* are admirable, as proving many interesting facts. But when studied with reference to the exact effect of jacketing, they present many inconsistencies which make it permissible to discount the apparent gain due the jacket as shown by the figures published, to the extent of a considerable fraction of this gain. For example, in one set of four experiments, Nos. 29 to 32 ("Peabody's Thermodynamics," page 278), it is possible to conclude that the jackets produced a loss of upwards of 12 per cent.

Again, we have a right to ask why the steam consumption of this engine for about one-quarter to one-sixth cut-off was four or five lbs. per H. P. higher than the amount which practice has clearly settled belongs to the conditions. This remark applies to the cases where the condenser was or was not used. The very best results, as shown by experiments 40 and 41, to be attainable only with the use of a jacket, are matters of every-day guarantee with first-class engines under similar conditions, but unjacketed. It is unfortunate that some of these experiments were not duplicated to determine the probable error of each test.

As regards the amount of superheating, it is an omission not to have stated it in my paper. It was carefully determined to be $2\frac{1}{2}$ degrees Fahrenheit. I am not aware that we have any right to believe that cylinder condensation during admission would be annulled by less than some 175 degrees of superheating. The amount of steam condensed during admission was 23 per cent. of the total steam consumption with the jacket in use, and about 28 per cent. without it.

As to giving all of the data of the test, I am very glad to do so, and it will be found in the form of an appendix to the paper.

There would be no essential difference in my conclusions resulting from using the water pressure as the mea-ure of work performed. But as the friction of the engine was variable, in consecutive tests over a possible range of about 2 per cent. of the total power, the same element of error presents itself. Regarding the finding of some 7 per cent. gain by dispensing with jackets on a certain pumping engine, I understand this engine to have run at about 15 revolutions per minute. I do not attempt to deny that at this speed of revolution there may be the amount of gain stated by the speaker; but I hardly think that his figures can be accepted as not subject to a possible correction, on account of the very short interval of time during which the jacket was out of action. Mr. Hague has informed me that the engine to which he referred was the Allis Compound Pump, at Allegheny, Pa. This engine has one high and two low cylinders, and expands steam about sixteen times at about 100 lbs. boiler pressure.

It is known that the radiation of a steam-cylinder is only about one per cent. of the steam used, estimated in heat, and any variation of this loss due to the presence of the jacket when not containing steam is not a sensible one in the problem.

I understand the engine referred to by Mr. Henthorn was built upon the same model as the Pawtucket engine tested by me, but was of somewhat smaller dimensions. I can only account for his facts on the ground that, in all probability, the steam supplied to this engine was superheated to the full extent which Mr. Corliss designed to have his boilers secure, namely, about 75 degrees, the experiments being a series carried on at Mr. Corliss' works previous to his making the Pawtucket engine contract. Mr. Corliss once informed me, during a conversation regarding the value of jackets, that he considered 5 per cent. of gain all which could be relied upon from their influence, and I have been assured at another time by his brother, Mr. William Corliss, that it is difficult to recall instances in the extensive practice of himself and his brother where there was *proof* of even this small amount of gain being attributable to jacketing. It should be borne in mind that this practice applies to engines making upwards of 50 revolutions per minute. I am glad to have the support of

so rigorous a student of physics as Professor Webb regarding the absence of any inconsistency between my position and the thermo-dynamic views which he has applied to the question of the usefulness of jackets.

I have found that it was the opinion among many practical American engineers that, whatever the gain by the use of the jacket, it was too small to make itself felt by the amount of coal burned, and in several instances the use of steam in jackets has been abandoned and the engines found to do their work with no apparent difference of action whatever. I do not mean to be understood as claiming that the jacket is not valuable on large marine engines for the purpose of heating up the cylinders after the engines have been idle, and in maintaining the cylinders freer from strains due to temperature. Many practical engineers believe in the value of jackets for this purpose, and their application to the latest types of marine engines must be regarded as valuable on these grounds. But so far as experimental proof of large economies due to jackets is concerned, this general criticism is to be made: First, some considerable change of conditions has generally existed between the tests with the jacket and those without; or, second, the economy gotten by the use of the jacket could be excelled or equalled by engines without a jacket, running under practically the same conditions. For example, Hirn's claim of 20 per cent. gain, based upon experiments made about 1859, is vitiated by the fact that the ratio of expansion with the jackets was about 9, and only about 14 without jackets. This range of expansion is now known to involve losses due to increase of cylinder condensation sufficient to discount greatly the claim for the jackets.

Hallauer's test on apparently the same engines, made several years later, finds but 8 per cent. gain in favor of the jacket, and his conditions involve a difference of boiler pressure of about 10 lbs. in 60, and one unit in the ratio of expansion, which was about 7.†

A similar analysis of Mr. J. G. Mair's * admirable tests to determine influence of jackets, and of the noted example in the *Bulletin* of the Mullhouse Society, 1878, where apparently 30 per cent. of gain is derived from the jacket, either leaves room to discount the claim of such gain on account of wide differences of

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* *Trans. Inst. Civil Engrs. of Gt. Britain*, Vols. LXX. and LXXIX. † See in *Smith's Steam Engine Practice*, the account of "Hallauer's Engine Tests."

condition, or to raise the point that nearly equal economy to that found by the use of the jacket is available by the best engines in general practice.

I do not mean to be understood as believing that all gain claimed for jackets by these experiments is delusive. Mr. Mair's tests on engines of slow rotational speeds afford a very good basis for the belief of considerable gains from jackets under such conditions. Regarding the gain held to be obtainable by jackets in Watt's time, his opinion was doubtless based upon the performance of the pumping engines. In these the range of temperature in the cylinder, by the principles of the equilibrium valve, was small; there was considerable wire-drawing which tended to keep the steam dry during admission, and at the same time cause a superior temperature to exist in the jacket, which makes it quite reasonable that 5 per cent. of steam spent in a jacket should have enabled cylinder condensation to be entirely avoided, and thereby some 20 per cent. of gain made available. This view is supported by the water consumption tests of Wicksteed on a Boulton & Watt engine. We may freely admit, therefore, that Watt's estimate of the jacket was well founded, and yet fail to realize any such gain in modern engines, because the latter do not realize the conditions of steam distribution of the Watt pumping engine.

I regret that one of the speakers does not give some explanation as to what *are* the conditions which permit a jacket to make so widely different a saving in different modern engines. Possibly his view is based on the theory of Cotterill; an ingenious view, but lacking probability in view of its ascribing to liquid water the ability to absorb heat with an improbable degree of rapidity.

The case cited by another speaker as occurring on one of the Cromwell Line steamers represents the opinion common among New York steam-boat men.

Regarding the experiments of Major English, I have not had time to study them closely; but in tests made by the writer and Prof. D. S. Jacobus * on a 17 x 30 engine, not having Corliss valves, the economy obtained was such as to make it questionable whether Major English's opinion regarding the special value of Corliss valves will be found supported in practice.

* "Steam Consumption at Variable Speed," *Trans. Am. Soc. Mechanical Engineers*, Vol. X., No. CCCXLIX., p. 722.

CCLXXV.

ON THE PERFORMANCE OF A DOUBLE-SCREW FERRY-BOAT.

BY E. A. STEVENS (ASSOCIATE) AND J. E. DENTON (MEMBER), HOBOKEN, N. J.

INTRODUCTION, BY E. A. STEVENS.

THE first propeller-boat used for ferry purposes was constructed in the first decade of this century by my grandfather, John Stevens, and made a run between Hoboken and Barclay Street, my uncles, John C. Stevens and Robert L. Stevens, acting respectively as pilot and engineer.

The engines of this vessel are at present in the Stevens Institute; and while the vessel would hardly be classed as a ferry-boat, in our understanding of the word, it is a curious coincidence that she was run over the very route on which the *Bergen* is now serving.

About forty years ago my uncle, Robert L. Stevens, and my father, E. A. Stevens, went so far as to have an estimate made by Hogg & Delamater, predecessors of the Delamater Iron Works, for a screw ferry-boat for the Hoboken ferries.

In 1867 a patent was obtained by Edwin L. Brady, of New York, for a screw-propeller ferry-boat. Two vessels of 900 tons burden were built under this patent. If they were used as ferry-boats at all it was to a very limited extent. They were used subsequently at the mouth of the Mississippi River as agitating dredges. It is believed that the washing of the levees caused by the quick water from the screws was so serious as to cause their use as ferry-boats to be abandoned.

Some twenty years ago Mr. Brady consulted on the matter of screw ferry-boats with the late Captain Woolsey, of Jersey City ferries, General McClellan, and Mr. William W. Shippen, then President of the Hoboken Land and Improvement Company. It was Mr. Brady's idea at the time that boats could be built under proper conditions.

About the same time, it is said, a single-screw vessel was used, with only partial success, on the Connecticut River, to transfer cars across that stream.

About, if not at, the same time, Mr. Francis B. Stevens, of Hoboken, made a model for a double-ended propeller ferry-boat, to which I will refer later, the subject having been considerably discussed by the management and by Professor R. H. Thurston, then of Stevens Institute and now of Cornell University.

In August, 1879, the *Oxton*, a double-ended boat with twin screws at each end, was placed in service on the Mersey, between Birkenhead and Liverpool. Since that time a number of similar vessels have been built and operated on the same route. The landings are made from the side of the vessels and not over her ends, as is the practice in this country. The vessels are considered successful, having great manœuvring power, and being more economical than the side-wheel vessels which they replaced.

Four years ago a paper was read before this Society in Boston by Mr. William Cowles, of New York, containing general drawings of a proposed screw ferry-boat, and comparing it closely with the prevailing type of ferry-boat in use in the New York harbor, and with an improved compound side-wheel boat suggested by him.

Mr. Cowles proposed using a toggle-joint on each side of his engine, so as to give proper submersion for his screws, which he further proposed to protect from ice by guard braces, and by a false stem projecting down in front and connected with a shoe running from the keel. He further proposed using a double smoke-stack, carried up on the divisions between the cabins and the team gangway.

The ferry across the Straits of Mackinac has long used propeller-boats, being considered by the company superior for use in ice, which in that locality often occurs in fields four feet in thickness, through which the vessels have to force their way.

In December, 1887, the steamer *St. Ignace* was launched from the works of the Detroit Dry Dock Company at Detroit. She was designed by Mr. Frank Kirby; is built of wood of great strength. Her general dimensions are 250 feet in length, 50 feet in beam, and 22 feet depth of hold. She has two compound engines, each one driving a propeller at opposite ends of the

boat; the forward engine propeller being small and less powerful than the after one.

It was found by experience that propeller vessels made better headway through heavy ice by proceeding stern first than by going ahead. The usual practice had been to back a vessel into a field of ice, run out ice anchors at each quarter as far as possible, then to start the vessel ahead, throwing the quick water from the propeller against the ice through which it was sought to force a way; to reverse the engine and back the vessel through the field which had been previously weakened by the flow of quick water.

Mr. Kirby's idea was that a forward auxiliary screw could be used to project a stream of water ahead of the vessel, so as to allow her to proceed continuously through ice of almost any thickness, the propelling force of the vessel in that case being the difference between the powers of the two engines. The idea in practice has been found entirely successful, the boat making much better time than any of the other vessels used in crossing the Straits.

The problem of constructing a screw ferry-boat has been a long-standing one with the Hoboken ferries. Early in the 70's, as previously noted, Mr. Francis B. Stevens, of Hoboken, got up a model and some preliminary drawings for such a vessel. The management, though not prepared for so radical a departure, kept the question before their minds as a possibility. Early in 1885 it became evident that two new boats must soon be built, and the question was raised whether they should be made propeller-boats or not. With some reluctance it was decided that there was not sufficient time to mature the necessary plans, as it became evident that the subject needed careful and close study. At that time, in connection with the Superintendent of Ferries, Captain C. W. Woolsey, I began a series of calculations and experiments which ended some two years later in the model of the *Bergen*.

Shortly after we had entered on the subject, Mr. Cowles' paper was read before this Society. The service demanded of a New York ferry-boat calls for some peculiar features of construction.

The weight of the loads carried, both in passengers and teams, as well as the strain caused by the ice, and the danger of collision, all call for a hull of great strength and rigidity. Beyond this, the vessel must have great stability, to resist burying by the

head as well as heeling. She must be able in floating ice, and should attain a speed of about twelve miles an hour in service.

The main characteristics of the *Bergen's* model are a full flar-

FIG. 71.

ing upper body, fine under-water body, with a full water-line, a sharp V-shape midship section (Figs. 71 and 72), and the peculiar cutting away of the ends to bring the rudders and screw within the perpendicular of the stems (Fig. 73).

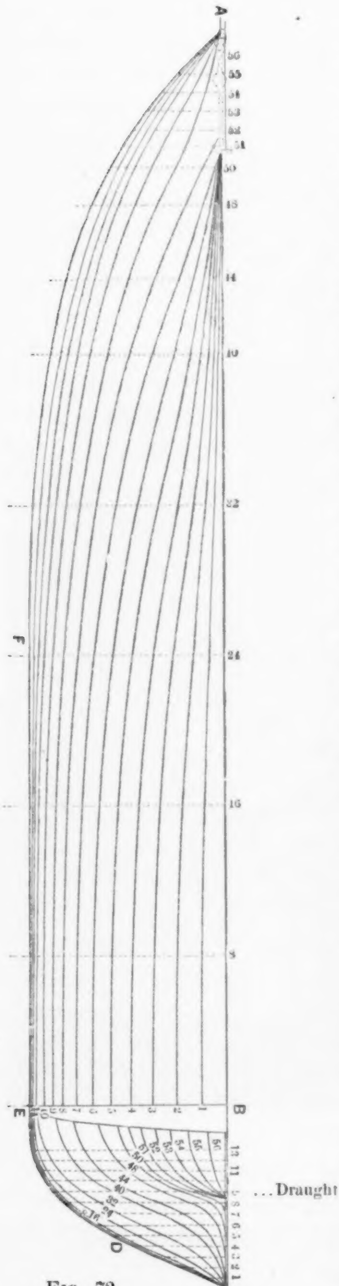


Fig. 72

The shape of the water-lines and upper body were determined by consideration of power in ice and stability.

The midship section, in order to give an unbroken line for the shafting, had to have a certain depth. It was found that with the required displacement the form adopted (Fig. 72) was about the only practicable one. The experience on the Hoboken ferry, moreover, had been very favorable to a sharp V section. The older wooden boats, built on a perfect V section, gave excellent results, while of the iron and steel boats, the *Orange* and *Montclair*, which more closely approach that section, gave better results than the other boats which had a semicircular section; and, as far as could be judged, than the West Shore and Pennsylvania R. R. ferryboats, with straight sides and a flat floor.

The experience on the Hoboken ferry, with balanced rudders hung under the keel and supported from above, having been very favorable both as regards efficiency, strength, and ease of repair, it was decided to use a rudder as nearly similar to the ones in use as conditions would allow.

The form shown in Fig. 73, with the ends of rudder stock supported in the shoe, appeared most suitable, and the fixed rudder in the bow seemed to promise protection from ice to the forward screw.

This arrangement has shown itself perfectly satisfactory in practice.

The question as to motive power presented four alternatives :

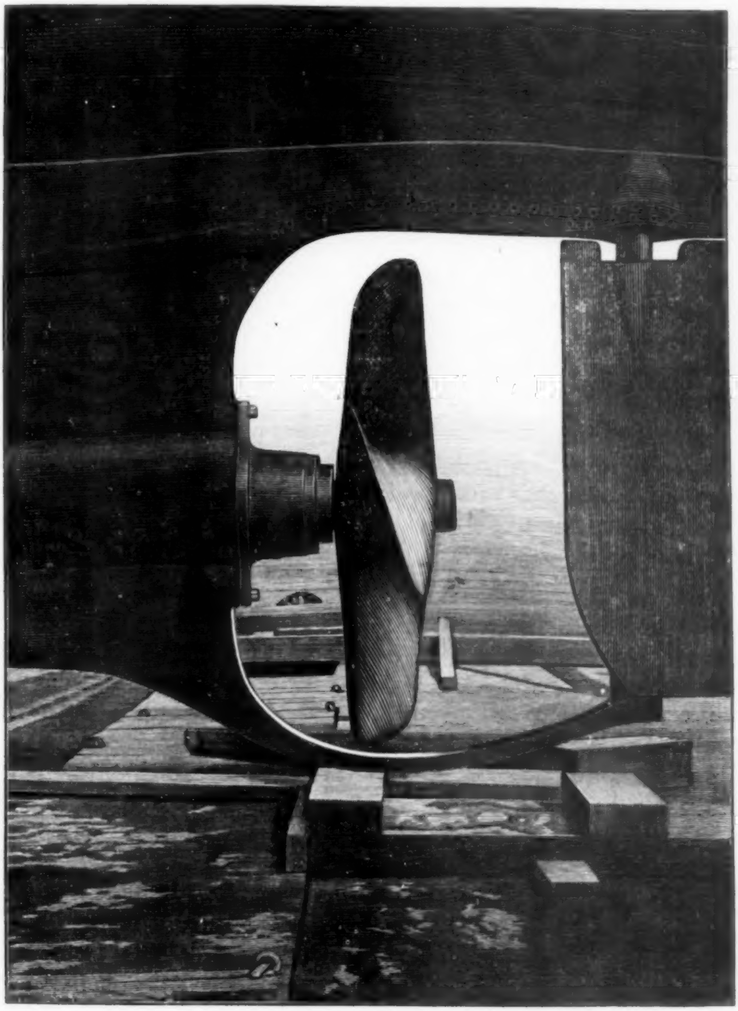


FIG. 73.

1. One engine, driving a line of shaft with universal flexible joints, there being an angle in the shaft on each side of the engine, as proposed by Mr. Cowles.

2. One engine, without such joints, and with a straight shaft, as in the *Bergen*.

3. Two engines, each driving independent shafts at an angle to each other, as in the *St. Ignace*, and, I believe, in the boats built by Brady.

4. Two engines, either with or without flexible joints in their shafts, driving two propellers at each end, as in the Mersey boats.

This last plan was rejected on account of the lesser protection from ice afforded the screws, and the fear of trouble in riding up on racks, as is often done when entering a slip with a strong wind and tide.

The advantages of the first and third methods were a deeper submersion of the screw and a flatter midship section. The disadvantages were the insecurity of the flexible joints in the first method; and in the third, the increased cost of construction and operation.

The cost of construction on the plan adopted has shown an advantage in the case of the screw. The hull is slightly more expensive, the engine decidedly less so, with the same power.

The engines and boilers of the *Orange*, 500 I. H.P., cost about the same as the *Bergen's*, 660 H.P.

I might add here that the *Bergen* is, if anything, more powerful than necessary.

In cost of operation it was found that, in a six weeks' run, the saving in coal would more than pay for the wages of an oiler and the extra lubricators used. Whether an oiler is necessary is doubtful.

As to capacity, the following table shows a comparison between the *Bergen*, *Orange*, and *Moonachie*.

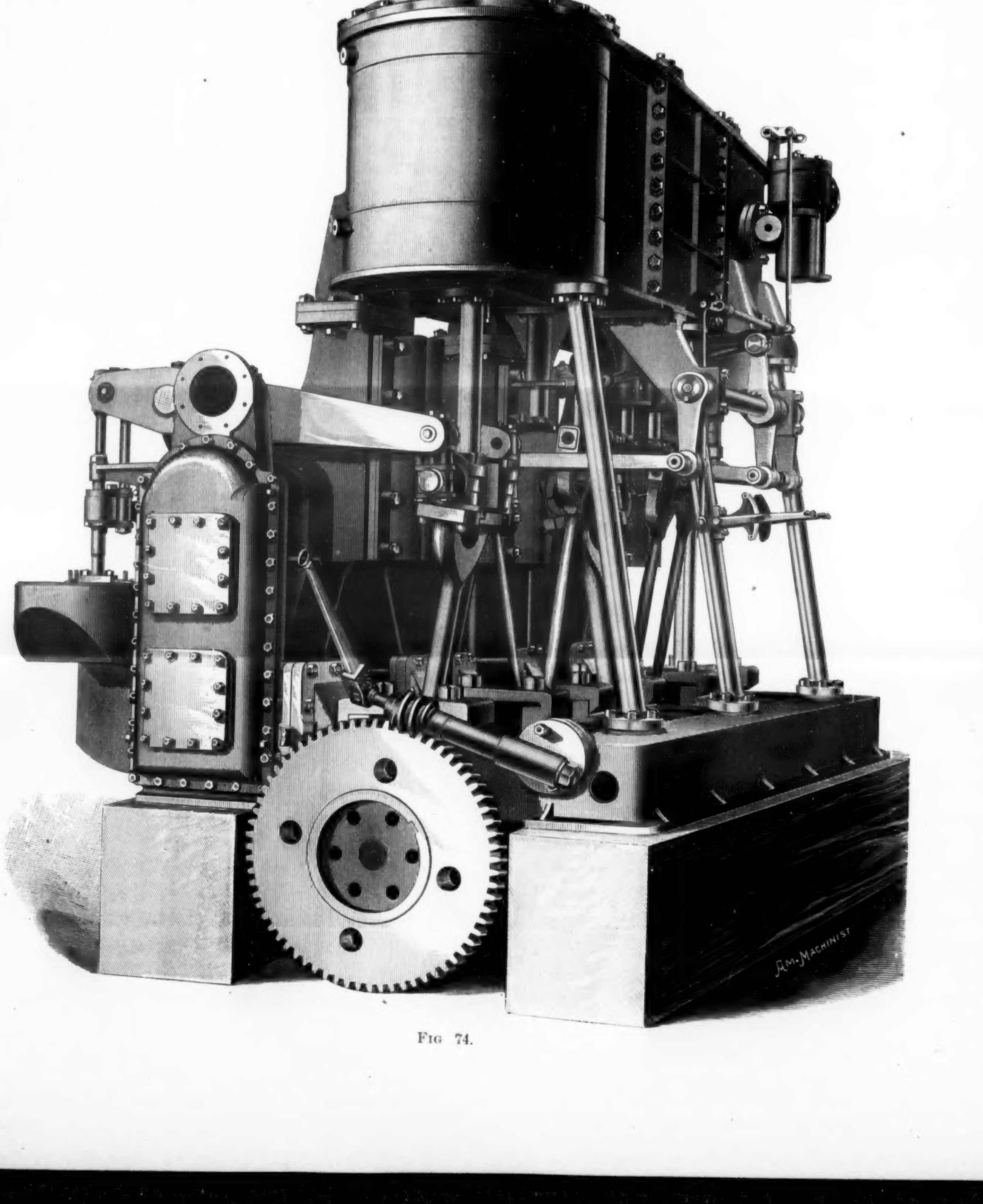


FIG 74.

	Bergen.	Orange.	Moonachie.
Built	1889.	1887.	1877.
Hull	Steel.	Steel.	Wood.
Engine, Type	Triple Expan. Propeller.	Low-pressure Beam.	Low-pressure Beam.
Size	$18\frac{1}{2}''$, $27''$, and $42'' \times 2'$.	$46'' \times 10'$.	$44'' \times 10$.
Safety-valve Pressure—Lbs. per sq. inch	160	45	30
Length, L. W. L.—Feet	200	217	200
Beam, " "	32.16	32	32
Beam, Over Guards "	62	62	62
Draught—Hull to Base Line	8.83	7.66	8
Displacement to L. W. L.—Tons	560	655	550
" per inch at L. W. L.	12.6	....	....
Space available for Pass.—Sq. Ft.	4,330	3,791	3,335
" " Number of Seats.	246	254	236
Space available for Teams—Sq. Ft.	3,448	3,940	3,380

It is to be noted in case of the *Bergen* that the space is more available, especially as regards teams, her team capacity being greater than the *Moonachie's*, and almost equal to the *Orange's*.

A decided advantage in the *Bergen* is the ease with which she can be unloaded, the saving in the crowded hours of travel being very considerable.

In point of handling, the *Bergen* compares very favorably with any ferry-boat on the river. Her greater draught makes her exceptionally steady on her helm, while it is found that she can turn as readily as other boats.

She can stop in a shorter distance, notwithstanding her higher speed.

The defects which have been found are not incident to the design, and while there are many departures that will be made in the engine in the next boat built, the hull will be practically unchanged, and no alteration in the engine involving any change in principle has as yet been found advisable.

In closing, I would say that the *Bergen* should be regarded as an experiment in adapting a style of propulsion which has been successfully employed elsewhere to the purposes of a ferry-boat in New York harbor. Whether use in heavy ice will show any defects which may require modification is as yet an unsettled question; but we feel full confidence that no more serious trouble than a change in the type of the propellers will be met.

Practically, the *Bergen* is preferred by passengers and pilots

alike. While the boat is by no means perfect, she is the best boat we have, and will furnish a type for our future boats.

The engines of the *Bergen* were designed by Mr. J. Shields Wilson, of Philadelphia. A number of engineers gave us advice and encouraged us by their confidence in the plan of building such a boat. Among the latter I may mention Mr. Frank Kirby, of Detroit, and Messrs. Herman Winter and Andrew Fletcher, of New York. A general view of the engine is shown in Fig. 74.

RESULTS OF EXPERIMENTS TO DETERMINE PERFORMANCE OF SCREW FERRY-BOAT "BERGEN."

BY J. E. DENTON.

The objects of the experiments undertaken were to determine the relative economy of the *Bergen*, as compared to the best type of paddle-wheel ferry-boat having the common style of overhead beam engine, a jet condenser, and drop-return flue boilers. The paddle-boat selected for this purpose was the *Orange*, one of a pair of steel boats designed in 1887 by Mr. Francis B. Stevens, and representing the best modern example of its class of ferry-boats. The programme carried out was as follows:

I. The steam consumption, boiler evaporation, horse-power, and speed were determined for each boat during 14 hours of regular ferry service.

II. Each was run to Newburgh and return, a distance of 120 miles, without stoppage, and the steam consumption per horse-power determined at the maximum capacity of the boilers. Also, the evaporative economy of the boilers, starting with new wood-fires, was determined during an interval of 14 hours, and the speed was measured by an estimate of the probable velocity of tides, and a log whose correction co-efficient was approximately known.

III. The speed of the *Bergen* was determined at the maximum horse-power for which the engines were designed, by opposite runs over a 1-mile course, after allowing the boiler pressure to accumulate above the average pressure which the boilers can maintain for more than a few minutes.

IV. One of the screws of the *Bergen* was removed, and the power and speed determined by runs over a 2-mile course, first with the single screw pushing, and then with it pulling, the boat at equal speeds of revolution of the engine.

The principal data and results are shown in Tables 1, 2, 3, and 4.

CONCLUSIONS.

The principal conclusions drawn from the experiments are as follows:

I. The steam used per horse-power for all purposes is 25 lbs. per hour for the beam engine, and 22 lbs. for the triple engine, under their average conditions of ferry service; but the consumption of the *Bergen's* main engine is only 18.3 lbs. per hour per H.P., the direct-acting steam feed and circulating pumps, etc., consuming about $3\frac{3}{4}$ lbs. per indicated horse-power.

II. The steam consumption of both engines does not practically differ while in intermittent ferry service from that found during continuous working of the engines. Pages 389 and 402.

III. The economy of the drop-return flue boiler of the *Orange* is practically the same as the locomotive type in the *Bergen*, both boilers evaporating on the average about $8\frac{1}{2}$ lbs. of water per lb. of bituminous coal, under ordinary working conditions, thus making the consumption of coal per hour per H.P. about 2.9 for the beam engine, 2.6 lbs. for the *Bergen*, for all purposes, and 2.15 lbs. for main engines alone.

IV. The speed of the boats under all conditions is practically in agreement with the law of cubes, and by the application of this law it appears that for a still-water speed of 12.6 statute miles per hour the following statements are practically true:

The paddle-wheel boat would require 642 H.P., and would make $24\frac{1}{2}$ revolutions per minute, with a slip of 26 per cent.

The screw boat, using double screws, would require 680 H.P., an engine speed of 145 revolutions, and the slip would be $12\frac{1}{2}$ per cent.

The screw boat, using one screw at the stern, would require 584 H.P., 152 revolutions per minute, and the slip would be 18 per cent.

The screw boat, using one screw at the bow, would require 692 H.P., 163 revolutions per minute, and the slip would be 18 per cent., but the recoil upon the hull of the water which the screw acts on would make the apparent slip about 22 per cent.

V. The screw at the bow, using the same horse-power as the screw at the stern for equal revolutions, propels the boat slower than the screw at the stern by an amount practically equal to

the equivalent of the extra resistance due to the increase of the velocity of the boat by an amount equal to the velocity of slip of the screw. See page 438.

VI. By calculations based upon the accepted relations between the slip of the screw and the velocity of a boat, it appears that, in order for the double screws to produce the same speed as a single screw of the same diameter at the stern, the slip of the latter must be to the former in the ratio of 18 to $11\frac{1}{2}$, and therefore the cause of the extra power consumed by the two screws as compared to the one screw is the fact that the slips are as 18 to 12.6, instead of as 18 to 11.5. The details of this calculation are given in the body of the paper.

TABLE I.

DIMENSIONS AND WEIGHTS OF "ORANGE" AND "BERGEN."

DIMENSIONS.	ORANGE.	BERGEN.
Length	211′	200′
Beam	32′	32′
Draught above base of hull	7′.8″	8′ 10″
Immersed surface	5,571 sq. ft	5,788 sq. ft.
Augmented surface	7,347 " "	7,524 " "
Co-efficient of augmentation	1.318	1.299
Mean angles of water lines	14°	13° 13′
WEIGHTS.	Lbs.	Lbs.
Boiler	76,000	100,852
Water in boiler	60,000—136,000	55,000—155,852
Propelling wheels	80,000	6,000
Propelling wheel shaft	24.000—104,000	27,000— 33,000
Engines without propelling wheels and shaft, but including frame, kelsons, etc.	132,000	122,000
Fresh water tanks filled, donkey pumps, attachments (piping, chimney, etc.)	80,000	86,000
Total machinery burden, exclusive of steering and ventilating engines	452,000	396,852
Hull as launched	860,000	720,348
Total weight, machinery and hull, as launched	658 tons.	558 tons.

TABLE II.

GENERAL SUMMARY OF EXPERIMENTS OF FERRY-BOATS "ORANGE" AND "BERGEN."

ENGINE.	ORANGE.	BERGEN. High.	BERGEN. Interm.	BERGEN. Low.
Diameter of Cylinder, inches	46 ins.	18½″	27″	42
Stroke, feet	10 ft.	2 ft.	2 ft.	2 ft.
Cut-off	0.45	[illegible]	[illegible]	[illegible]
Clearance	3.7%	16%	10.7%	11.3%
Total Expansion	2.1	....	9	.....
Area of Admission Ports, per cent. of Piston.		12%	12½%	11%

BOILERS.	ORANGE.	BERGEN.
Total Heating Surface, sq. ft.	3,049	3,462
Superheating " "	0	0
Grate Area, sq. ft.	80	81
Ratio of Grate to Heating Surface	38	43

PRESSURES.	ORANGE. LBS. ABOVE ATMOSPHERE. 120-Mile Run.	ORANGE. LBS. ABOVE ATMOSPHERE. Ferry Service.	BERGEN. LBS. ABOVE ATMOSPHERE. 120-Mile Run.	BERGEN. LBS. ABOVE ATMOSPHERE. Ferry Service.
Average Boiler Pressure	17	32	114	140
" Pressure During Admission	16	31	105	100
" Back-pressure		4½		3
" Vacuum Pressure		27 ins.		27 ins.

TEMPERATURES.	ORANGE. 120-Mile Run.	ORANGE. Ferry Service.	BERGEN. 120-Mile Run.	BERGEN. Ferry Service.
Feed-water	93°		118°	
Uptake	500°		750°	
Top of Stack	435°		650°	

TABLE II.—*Continued.*

GENERAL SUMMARY OF EXPERIMENTS OF FERRY-BOATS "ORANGE" AND "BERGEN."

Indicated Horse-power.	Orange.		Bergen.	
	120 Mile Run.	Ferry Service.	120 Mile Run.	Ferry Service.
I. H. P., including all pumps	490	810	....	....
I. H. P., not including pumps	....	....	665	650
I. H. P., feed-pump	....	....	9	....
I. H. P., circulating pump	....	....	3	....
I. H. P., bilge pump	....	....	1½	....
Total Weights.				
Bituminous coal per hour, lbs	1,560*	....	1,580*	....
Percentage of ashes	11%	....	7.87%	....
Feed-water per hour for all purposes	13,487	....	14,511	....
Feed-water per hour for pumps, etc	....	....	2,358	....
Feed-water per hour for steering engines, lbs	....	150	....	150
Efficiency of Boilers.				
Evaporation at actual pressure and temperature of feed per lb. of coal	8.65	....	9.2	8.42
Evaporation from and at 212° per lb. of combustible	11.00	....	11.40	....
Efficiency of Engine.				
Water for all purposes per hour per I. H. P.	27	25	21.8	22.9
Water, main engines only, per hour per I. H. P.	....	....	18.3	....
Water, feed, and circulating pumps, etc., per hour per I. H. P	....	....	160	130
Theoretical water per hour per I. H. P. } Calculated from card }	20	....	Condensing. 13.2	Non-condensing. ....

* These amounts are estimated from the feed-water consumed, by use of the figures for evaporation per lb. of coal, as determined from the boiler tests. See pages 386 and 392.

TABLE III.

SUMMARY OF SPEED DETERMINATIONS OF "BERGEN."

Conditions.	Revolutions per Minute.	Horse-power.	Observed Still-water or True Speed. Statute Miles per Hour.	Slip per centum.	Estimated Speeds. From a Speed at 145 Revs. by Law of Cubes.	From Augmented Surface. $V = \sqrt[3]{\frac{21{,}200\text{ H.P.}}{\text{Aug. Surf.}}}$	From Apparent Slip.
	1	2	3	4	5	6	7
Two Screws in use....	142	662	11.9	16.4	12.37	14.19	11.8
" " "	145	700	12.62	12.6	12.60	14.57	13.4
" " "	162	1,007	14.6	11.0	14.30	16.13	16.2
" " "	114	334	10.5	10.?	10.50	11.27	11.6
" " "	71	97	6.4	10.?	6.00	7.45	7.8
One Screw at Stern...	145	458	11.96	18.2	11.96	12.5	12.7
" " " ...	163	684	13.42	17.7	13.67	14.32	14.4
" " " ...	83	93	6.98	16.0	7.30	7.36	7.8
One Screw at Bow....	145	461	11.28	22.3*		11.99	12.0

* Assumed to be 18 per cent. for calculation of Column 7.

TABLE IV.

SPEED OF PADDLE-WHEEL BOAT "ORANGE."

Revs.	H. P.	Observed True Speed. Statute Miles.	Speed Calculated from Aug. Surface. $V = \sqrt[3]{\frac{18{,}000\text{ H. P.}}{\text{aug. surf.}}}$	Apparent Slip. Percentage.	Speed Estimated by Law of Cubes. Statute miles.
22.9	490	11½	12.1	26	....
24.6	642	....	....	26	12.6

TABLE V.

MEASUREMENTS MADE ON SCREW OF "BERGEN," AS PER FIG. 75.

Angle between lines in Col. 1.		Points between which measurements are taken.	Vertical Distances from plane of Hub face to point.	Pitch.	Angle between lines in Col. 1.		Points between which measurements are taken.	Vertical Distance from plane of Hub face to point.	Pitch.
Designation.	In Degrees.				Designation.	In Degrees.			
1-0-11	22° 14′	19-21	$6\frac{1}{16}$	8.184	22-0-23	7°10′	22-23	$5\frac{7}{8}$	24.49
		14-16	$6\frac{15}{16}$	9.366	28-0-18	3°35′	28-18	$2\frac{13}{16}$	23.51
		7-9	$6\frac{13}{16}$	9.197	27-0-12	1°12′	27-12	1	25.00
		1-10	$6\frac{1}{2}$	8.756					
1-0-5	38° 18′	19-22	$10\frac{1}{4}$	8.029					
		14-17	$11\frac{5}{16}$	8.861					
		7-13	$11\frac{7}{16}$	8.959					
		2-4	$11\frac{5}{8}$	8.5					
		3-5	$9\frac{1}{4}$	7.245					
3-0-11	18° 52′	20-21	$4\frac{9}{16}$	7.254					
		15-16	$5\frac{1}{8}$	8.148					
		8-9	$4\frac{7}{8}$	7.750					
		2-10	$4\frac{3}{4}$	7.552					
		3-11	$4\frac{11}{16}$	7.452					
11-0-5	15° 34′	21-30	$4\frac{5}{8}$	8.912					
		16-17	$4\frac{3}{8}$	8.431					
		9-13	$4\frac{5}{8}$	8.912					
		10-4	$4\frac{9}{16}$	8.791					
		11-5	$4\frac{9}{16}$	8.791					
3-0-5	31° 44′	3-5	$9\frac{1}{4}$	8.744					
1-0-6	45° 20′	19-22	$12\frac{7}{16}$	8.229					
		14-28	$13\frac{9}{16}$	8.974					
		7-27	$13\frac{11}{16}$	9.057					
		1-6	$13\frac{1}{2}$	8.932					

Average pitch for Pushing side of blade for points, 3-5, 7-27, and 1-6	8.911
Average pitch for Pulling side of blade for the same points	8.920
Projected area of blades in per cent. of disk area	53.1
Area of boss	3.5

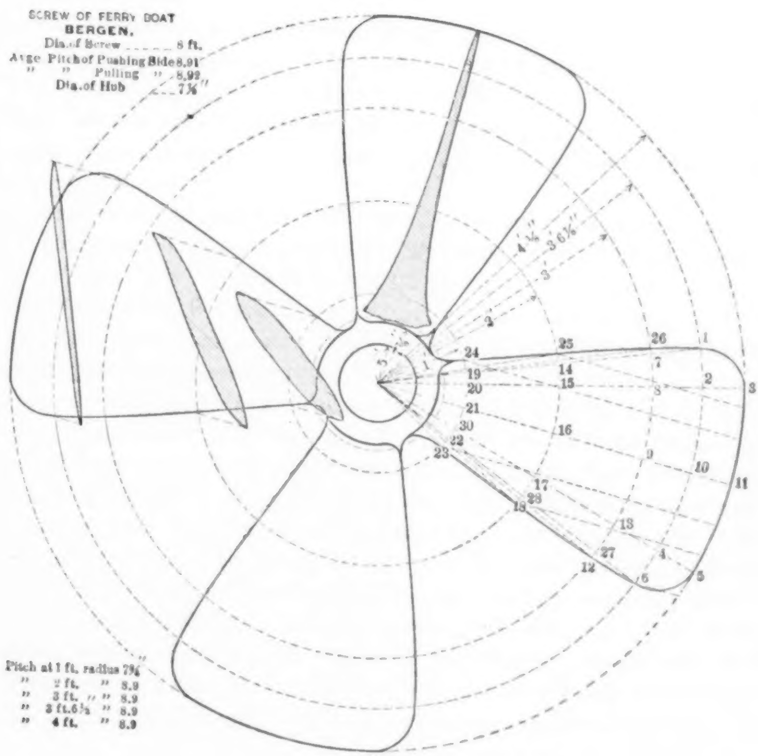

SCREW OF FERRY BOAT
BERGEN.
Dia. of Screw 8 ft.
Av'ge Pitch of Pushing Side 8.91
" " Pulling " 8.92
Pitch at 1 ft. radius
" 2 ft. " 8.9
" 3 ft. " " 8.9
" 3 ft. 6½ " 8.9
" 4 ft. " 8.9

Fig. 75.

PART I.

EXPERIMENTS WITH PADDLE FERRY-BOAT "ORANGE."

ONE HUNDRED AND TWENTY MILE TEST OF PADDLE-WHEEL FERRY-BOAT "ORANGE."

With the boat at the dock, the banked fires were spread at 4 A.M., Aug. 15, and steam raised to 20 lbs. The engine was then worked until steam fell to 10 lbs. At 6 A. M. the fire was hauled, the grates and ash pit being well cleaned.

A new fire was then made, using 1,425 lbs. of cord-wood. This fire was lighted at 6.20 A.M., and the boiler evaporation test dates from this time.

The steam pressure was then 8 lbs., and the boiler contained about 81,000 lbs. of water. The steam pressure commenced to rise at 7 A.M., and was about 20 lbs. at 8 A.M.

The engine was worked at the dock until 9 A.M., when the boat left for Newburgh. The throttle was opened wide at 10.30, and the water consumption test of the engine alone dates from this time, a special mark being made on the water glass to note the water-level in the boiler. From the time of starting up to 10.30, the engine being under throttle, a boiler pressure of 30 lbs. was maintained, and the cards taken at 10 25 showed 496 H.P., the speed of the engine being 704 revolutions in 30 minutes, or 23.5 per minute. When the throttle was opened wide, the boiler pressure quickly fell to 20 lbs., but during the run from 10.30 to 11 A.M., 558 H.P. were indicated, and the engine made 719 revolutions, or 23.9 per minute. The average boiler pressure was 19 lbs. for this interval. Steam continued to fall up to 12 A.M., when the pressure was 17 lbs. The H.P. averaged 482 and revolutions 23.0 per minute.

The steam pressure averaged 23 lbs. until the arrival at Newburgh, about 1.52 P.M. The H.P. for this interval averaged 581, and the revolutions 24 per minute. The engine was checked at Newburgh only long enough to reverse the direction of its rotation, about three minutes being consumed before the boat gained full headway toward New York.

From causes not determined, the steaming capacity of the boilers rapidly decreased after 2 o'clock, the steam pressure being

$14\frac{1}{2}$ lbs. at 2.30 P.M., and $11\frac{1}{2}$ at 3.30, with the revolutions at 21.8 per minute and H.P. 420. The engine was then throttled to allow steam to rise, and at 4 P.M. the boiler pressure was 17 lbs., the revolutions being 20.3 for the half-hour and the H.P. 376. Steam rose steadily to 24 lbs. at 4.30 and 30 at 5 P.M., and averaged 27 lbs. until 6.30 P.M., when the boat reached her dock at Hoboken. The amount of throttling is shown by cards *a* and *c* (Fig. 83). The speed during this interval of three hours under throttle averaged 22.5 revolutions per minute, and the H.P. averaged 444. The amount of steam used during the five hours under full boiler pressure was determined by a meter to be 1,095 cubic feet. Each cubic foot registered by the meter was determined by calibration of the latter (under its actual working conditions) to represent 65 lbs. Hence, in five hours, 71,175 lbs. of steam were generated, or 14,235 lbs. per hour.

A 7″ x 9″ * Niagara direct-acting steam-pump used 233 lbs. per hour from 12 o'clock until 4.30. Hence the engine alone used $14.235 - \frac{3}{5} \times 233$, or 13,996 lbs

The average H.P. from 10.30 A.M. to 3.30 P.M. was 518. Hence the steam used per hour per H.P. was 27.25 lbs.

The amount of steam used during the three hours under throttle was 36,725 lbs., from which is to be subtracted one hour's use of the pump, making 36,492 lbs. of steam for three hours, or 12,164 lbs. per hour. The average H.P. was 444, making the consumption of steam per hour per H.P. 27.40 lbs. In using the level of the water in the boiler as a basis for estimating the heating of the water contained therein, an error of $\frac{1}{2}$″ of level may enter at either the beginning or end of the test. This involves 1,200 lbs. of water as a possible error due to 1″ of difference of water-level. For the five-hour test the resulting variation in steam consumption per hour per H.P. is $\frac{1,200}{5 \times 500} = 0.5$ lbs., and for the three-hour test $\frac{1,200}{3 \times 444} = 0.9$ lbs.

The possible variation due to the estimation of the pump's

* This pump delivered water from the river into the feed-tanks; it was run under throttle at probably 10 lbs. pressure above the atmosphere, at 100 revolutions per minute. It was started at 12 o'clock and stopped at 4.30. Its steam consumption is estimated assuming 10 per cent. clearance, and that three times as much steam entered the cylinder as would fill its piston and clearance volume. The error in this computation does not exceed one-third of the steam involved for pump alone, or two-thirds of one per cent. of the steam used by the engine.

consumption of steam is about 0.2 lbs. per hour per H.P., and a further variation of 1 per cent., or 0.2 lbs., may arise from the possible error of calibration of the meter. It therefore follows that the measurement here recorded determines the consumption of the engine between the following limits only:

	Possible Minimum.		Possible Maximum.
Steam per hour per H.P. without throttle	26.25	to	28.25
" " " using "	26.00	"	29.00

Manifestly no deduction regarding the relative economy of throttle *vs.* no throttle should be made from these figures. The most reliable deduction is that under mixed conditions of running with part-open and full-open throttle, such as commonly occur in practice, the consumption per hour per H.P. averaged, for the entire eight hours, 27.3 lbs. of steam. Table I. gives details of observations during the trip.

TABLE VI.

GENERAL OBSERVATIONS, 120-MILE TEST PADDLE-WHEEL BOAT "ORANGE," AUGUST 15, 1889.

Time.	Steam Pressure Gauge.	Vacuum inches.	Revolutions.		Horse-power.	Remarks.
			Reading of Revolution Counter.	Per Min.		
A.M.						
10.00	..	..	0	...	....	
10.30	18	..	704	23.5	498	
11.00	20	26	1,423	23.9	558	
11.30	17¼	26	2,120	23.0	482	
12.00	17	25¾	2,807	23.0	482	
12.30	24¾	26	3,526	23.9	603	
1.00	23	25¼	4,245	23.9	573.5	Throttle wide open. See Card *b*, Fig. 83.
1.30	22	25¼	4,975	24.3	568.5	
2.00	21	25½	5,618	24.3	568.5	
2.30	14½	26	6,343	24.2	520.5	
3.00	14¼	26	7,025	22.7	463	
3.30	11¾	26	7,679	21.8	419.5	
4.00	17¼	26	8,287	20.3	376.5	
4.30	24½	25¾	8,944	21.9	470.5	
5.00	30	26	9,619	22.5	478	
5.15				22.8	485.5	Throttle partly closed. See cards *a* and *c*, Fig. 83.
5.30	25¼	26	10,311	23.1	456.5	
5.45	21	26			433	
6.00			10,978	22.2	440	
6.20			11,424		454.7	
Average.	17	26		22.9	490	

The boiler test ended at 7.20 P.M., making it extend over an interval of thirteen hours. The pressure at the time of drawing the fires was 20 lbs. The total water fed was 130,260 lbs. The

total coal, 14,915 lbs. Taking the 1,425 lbs. of kindling wood as equivalent to $\frac{4}{10}$ as much coal, we have the total fuel equal to 15,235 lbs. of coal. The fire was well burned out at the close of the test. The total refuse from grates and ash pit was 1,659 lbs. As the water in the boiler was at 260° at the close of the test and at 240° at the start, we must credit the fuel with the heat to raise the weight of water in the boiler through 20° Fahrenheit. This amounts to $81,000 \times 20 = 1,620,000$ thermal units. The steam generated was determined to be perfectly dry by the jet

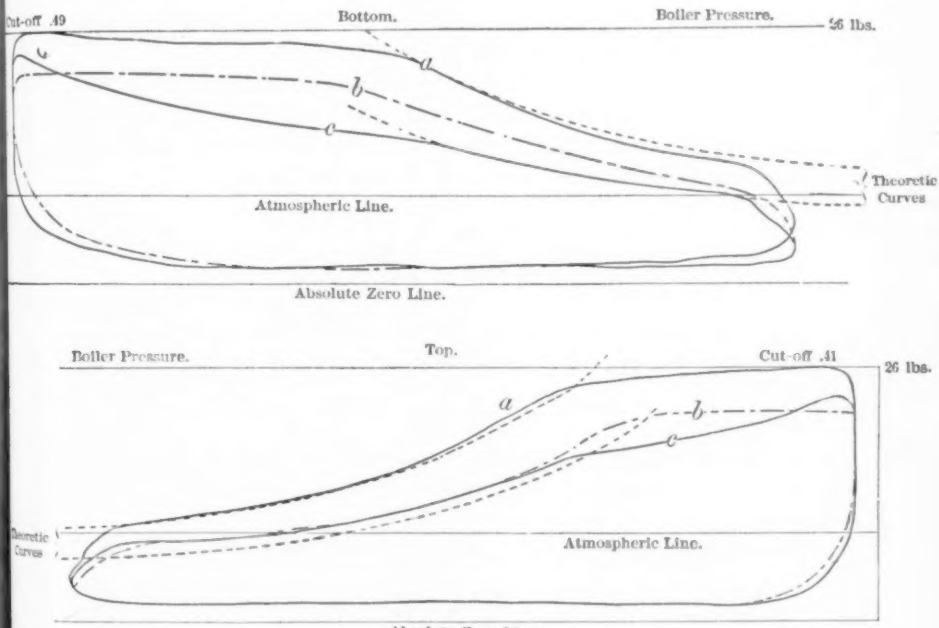


FIG. 83.—120-mile run of "Orange."

method. Its average pressure was 20 lbs. above the atmosphere, for which the total heat of evaporation is 1,192 thermal units. The average temperature of feed-water was 93° Fahrenheit. Hence each pound of steam required of the fuel $1,192 - 93 = 1,099$ thermal units.

Therefore, 1,620,000 thermal units represent the evaporation of 1,473 lbs. of water from 93° into steam at 20 lbs. The total evaporation of water was therefore 131,733 lbs., with 15,235 lbs. of Virginia bituminous coal, or 8.65 lbs. of water per pound of

coal. The ashes amount to 10.9 per cent. of the coal. The average temperature of uptake was 500°. If the air per pound of coal was 19 lbs., the heat escaping by chimney per pound of coal was 20 × (500° × 80° temp. fire room) × 0.24 specific heat, = 2,016 thermal units. Taking 13,000 heat units as the total heat evolved by the combustion of a pound of the coal with the above percentage of ashes, the waste by chimney is 15.4 per cent. The steam absorbs $\frac{8.65 \times 1,099}{13,000} \times 100 = 73.2$ per cent. This leaves 11.4 per cent. to be dissipated by radiation from the exterior of the boiler and ash-pit, and heat contained in the fuel and refuse on grates and in ash-pit at the close of the test.

If the evaporation had taken place under atmospheric pressure, and the feed-water been at 212° Fahrenheit, each pound of steam would have required 966 thermal units from the fuel. Hence the evaporation per pound of coal would be

$$8.65 \times \frac{1,099}{966} = 9.83 \text{ lbs.}$$

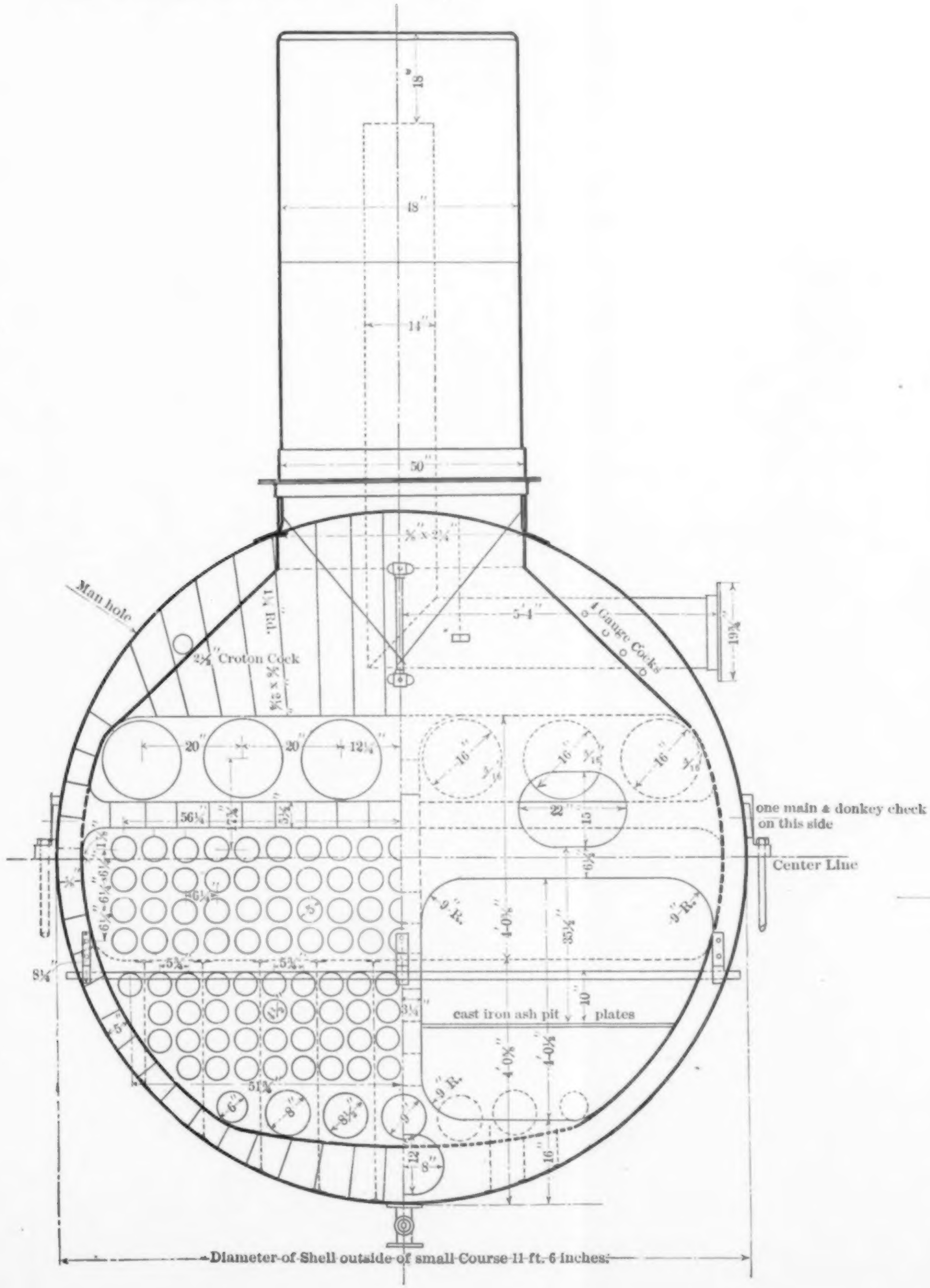
The total combustible being 89 per cent. of the total fuel, the evaporation per pound of combustible from and at 212° will be $\frac{9.83}{0.89} = 11$ lbs. Boilers of first-class design evaporate from 11 to 12 lbs. of water per pound of combustible from and at 212°, with natural draught. The boiler of the *Orange* is shown in Fig. 85.

The Heating Surface of the Boiler is distributed as follows:

Furnace	217 sq. ft.
6–16″ Flues	250 " "
76– 5″ Tubes and Rear Connection	1,088 " "
72– 4½″ Tubes	1,493 " "
2– 6″ "	
2– 8″ "	
2– 8½″ "	
1– 9″ " and Middle Connection	
Total	3,048 sq. ft.
Total Grate Surface	80 sq. ft.
Heating Surface divided by Grate	38 " "
Steam Space	500 cu. ft.

Approximate determinations of temperatures gave results as follows:

BOILER OF FERRY BOAT ORANGE.



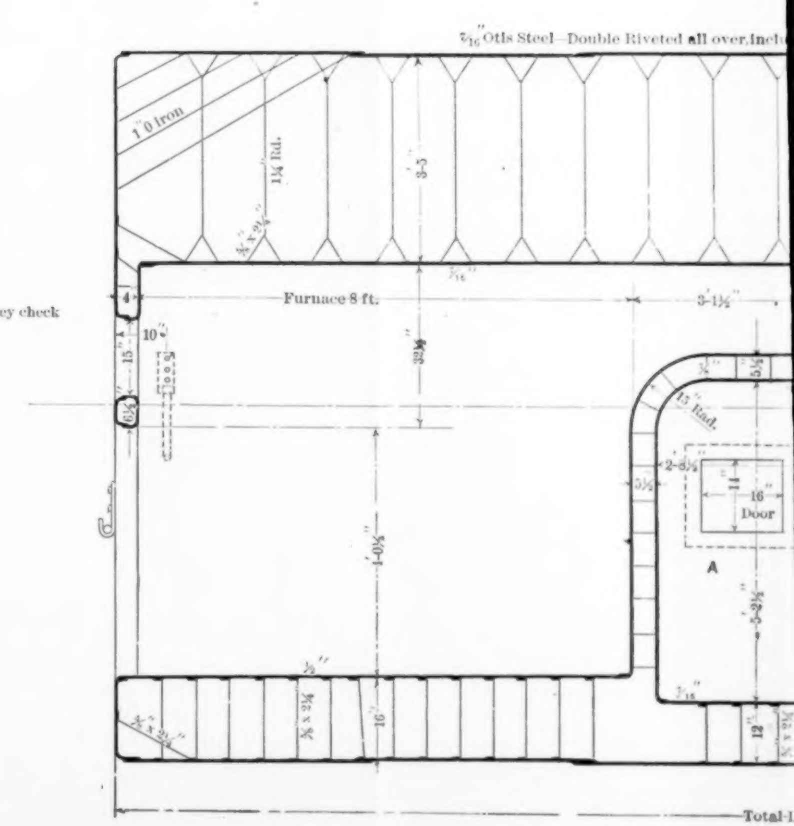


FIG. 85.

BOILER OF FERRY BOAT ORANGE.

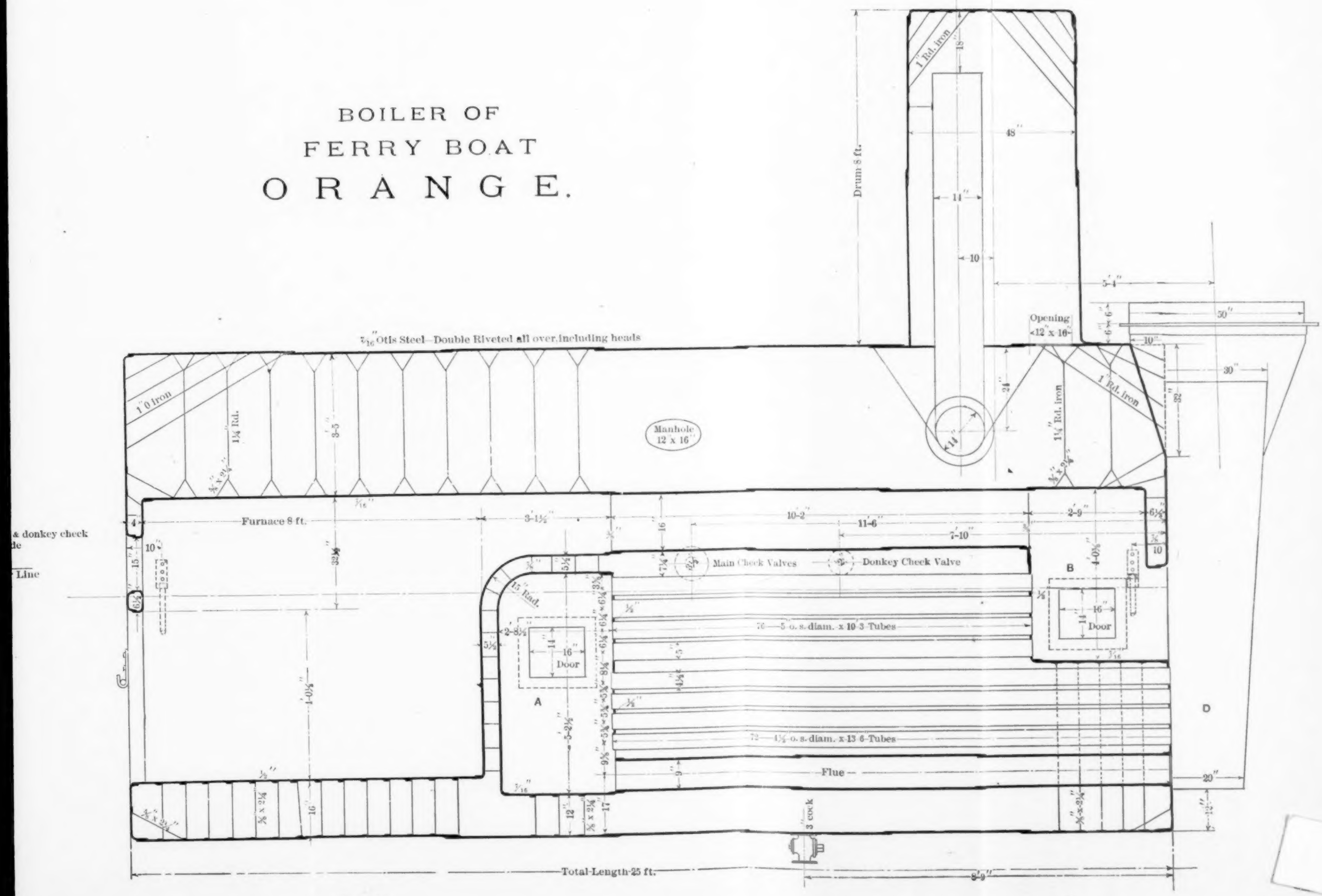


FIG. 85.

Temperature of Fire		2,500° Fahr.*
"	" Gases Leaving Furnace	1,600° *
"	" Rear Connection	1,000°
"	" Middle "	700°
"	" Uptake	500°
"	at Top of Chimney	435°

NOTES ON THE THEORETICAL DISTRIBUTION OF CHIMNEY DRAUGHT.

The chimney was of unprotected sheet-iron. Its diameter was 50 inches, and height above the top of the boiler 46 feet. The surface exposed to the atmosphere was, therefore, about 575 square feet. Péclet gives for the rate of loss of heat from such a surface 1.5 heat units per square foot per hour for each degree difference of temperature between the interior and exterior surfaces of the chimney.

According to this the loss of heat per hour would be $575 \times 1.5 \times 400 = 350{,}800$ heat units, assuming that the average difference of temperature was 400°.

The total heat lodged in the escaping gases at 500° temperature would be $500 \times 16 \times 0.24 \times 1{,}400 = 2{,}500{,}000$ heat units; the weight of gas per pound of coal being assumed at 15 † lbs., the weight of coal consumed per hour being 1,400, and the specific heat of the products of combustion being 0.24.

Consequently, the loss of heat in passing up chimney should be $\frac{350{,}800}{2{,}500{,}000} \times 500^\circ = 70^\circ$.

The actual loss of temperature observed was 65°.

The total height of chimney producing draught was 54 feet. The rate of combustion was $\frac{1{,}400}{80} = 17.5$ lbs. of coal per hour per square foot of grate.

The cross-section of chimney			is 15 sq. ft.	= 0.18	× grate surf.
" " "	1st set of flues	is	8.4 sq. ft.	= 0.095	× gr. sur.
" " "	2d " "		10.5 "	= 0.12	× "
" " "	3d " "		10.5 "	= 0.12	× "

The weight of gases passing through these sections per second, assuming 15 lbs. per pound of coal, is $\frac{1{,}560}{3{,}600} \times 15 = 6.5$ lbs. At the temperature of the chimney the volume per second will be

* Estimated from J. C. Hoadley's "Measurements on Warm Blast Boiler."

† This figure is per pound of coal, not carbon. It is assumed by comparison with the measurements made on the *Bergen's* draught.

$6.5 \times 25.0 = 163$ cubic feet; making the velocity up chimney $\frac{163}{15} = 10.8$ feet per second.

The frictional resistance of a stack 50 inches diameter, and 46 feet long, due to this velocity, is equal to the weight of a column of gas one foot square and of a length equal to $\frac{(10.7)^2}{2g} \times \frac{.05 \times 46}{\frac{50}{12}} = 1$ foot; in which g is the acceleration due to gravity, or 32.16, and .05 a co-efficient of friction for sooty surfaces. At the temperature of 500° the density of the chimney gas is practically one-half that of the atmospheric air, so that the difference between the weight of a column of such gas and a column of air of equal height is equal to the weight of the column of gas itself. This difference is the force producing the chimney draught. Hence, the 1 foot of frictional resistance due to the chimney may be considered to represent the effect due to 1 foot of the chimney's height. In passing through the six 6″ flues, the density of the gases is one-half that of the gases in chimney, but the velocity is thereby twice as great for equal area. The actual velocity for equal volume is $\frac{150}{8.4} = 19$ feet per second. Taking the friction inversely as the density and as the square of the velocity, we have the height of chimney to overcome the resistance, the flues being 10 feet long.

$$\frac{(19)^2}{2\,g} \times \frac{.05}{\frac{6}{12}} \times \frac{(2)^2}{2} \times 10 = 4 \text{ feet of chimney height.}$$

Similarly, for the resistance in passing through the 76-5″ tubes there is required 5.4 feet of chimney, and for the 72-4½″ tubes, etc., forming the last pass to the stack, there is required 4.6 feet of chimney.

To accelerate the gases from a state of rest to their final velocity in the chimney, requires 2 feet of chimney; and for the three changes of direction we should allow three times as much as for acceleration, or 6 feet of chimney. For the resistance to passage through the grate and fire we should, by Péclet and Rankine, allow $0.0017 \times$ Volume Gases flowing per second at temperature of 500° in cubic feet. This gives 38 feet.

Hence, the total height of chimney required should be,

	Feet.
For acceleration	1.8
" Chimney resistance	1.0

		Feet.
For Turns in passages		6
" Friction through passage of 16" flues		4
" " " " " 5" "		5.4
" " " " " 4" "		4.6
" " " grate and fire		.38
		60.8 ft.
	The actual height was	54.0 ft.

A draught gauge 46 feet below the top of the chimney averaged 0.33 inches of water. By formula, page 288, Rankine's St. Eng., we have

Head in inches of water $= 0.192 \frac{T_o}{T_1}\left(0.0807 + \frac{1}{Vo}\right) \times H$, Vo being the volume of air at 32° supplied to each lb. of fuel, and $To + T_1$ the absolute temperatures, and H the height to top of chimney. Taking Vo at 250, $To = 500$, and T_1 1,000°, we have head of water by theory equal to 0.37 inches. As the water gauge cannot be read as closely as $\frac{1}{32}''$, the agreement between theory and observation is quite satisfactory.

TEST OF PADDLE-WHEEL FERRY-BOAT "ORANGE" IN REGULAR SERVICE, MAKING HALF-HOURLY TRIPS FROM BARCLAY STREET TO HOBOKEN.

Commencing at 7 hrs., 02 mins., 30 secs., P.M., August 8, the performance of the boat was recorded until 9 hrs., 42 mins., 10 secs., A.M., August 9. Table VII. shows the detailed observations. Thus, in Column 1 is given the time of leaving the slip; in Column 2, the time when the engine was at full speed; Column 3, the interval between these events, or the time at which the engine was at half-speed; Column 4 gives the time when the engine was "slowed down;" Column 5, the period during which full speed had been maintained; Column 7 gives the period during which engine was worked with the bar; Column 8 gives the delay in river after leaving the slip; Columns 15 and 16 show how long steam wasted though the safety-valve; Column 18 gives the reading in cubic feet of the meter which measured the feed-water. Columns 19 and 20 give the revolutions made by the engine.

TABLE VII.

TRIAL OF "ORANGE" IN FERRY SERVICE ON HALF-HOUR TIME BETWEEN HOBOKEN AND BARCLAY STREET, NEW YORK CITY, AUGUST 8 AND 9, 1889.

	Full Speed.		Slow Down.		End of Trip.			Steam Pressure.			Vacuum.			Safety Valve Blowing Off.				Counter.	
Time.	Time.	Interval.	Time.	Interval.	Time.	Interval.	Delay.	When Leaving.	On Arrival.	Average.	When Leaving.	On Arrival.	Throttle.	Beginning.	End.	Revolutions per Minute.	Meter Readings.	Total.	Difference.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
H.M.S.	H. M. S.	Secs.	H. M. S.	M. S.	H. M. S.	M. S.	Secs.	Lbs.	Lbs.	Lbs.	Ins.	Inches.					Cu. Ft.		
7.02.30	7.02.45	15	7.13.8	10.15	7.15.30	2.30	20	40	22	31			0.4	7.21	7.22¼	24	2,258	0	
.22	.22.45	45	.29.30	6.45	.31.15	1.45	0	42	25	33.5			.5	.3740	.4045			217	217
.40.10	.40.30	20	.49.15	8.45	.51.40	2.25	0	42	30	36	26	26.5	.5			26.5		430	213
			8.01.20		8.03.45	2.25	0	39	30	34.5	26	27.5	.5			26		640	210
8.74. 5	8. 8	15	.17.20	9.20	.19.30	2.10	0	38	26	32	27	28	.55			25	2,355	856	216
.22.30	.23.20	50	.30.30	7.10	.32.30	2.	40	39	28	33.5	27	28	.55			26		1,073	217
.38.20	.38.40	20	.47	8.20	.49.45	2.45	0	40	21	30.5	27	28	.5			25		1,280	207
.52.30	.53.20	50	9.05.0	7.30	9.25.5	2.05	0	34	23	28.5	28	28	.5			25		1,481	201
9.17	9.17.20	20	.25	7.40	.27.45	2.45	0	38	25	31.5	27	28	.5			24		1,698	217
.31	..32.30	90	.39.30	7.	.42.15	2.45	10	35	26	30.5						27	2,621		
.46	.46.30	30	.54.30	8.	.57.30	3.		36	23	29.5			.5					1,906	208
10.02	10.02.40	40	10.11	8.20	10.13.30	2.30		42		30			.5	10.02	10.0210		2,770	2,173	267
.16.07	.17.30	83	.24	6.30	.26.40	2.40		36	26	31	27	27	.5			24		2,317	144
.31.05	.32	55	.39.30	7.30	.42.50	3.20		36	24	30	26	28	.5			26	2,784	2,534	217
.46.25	.46.40	15	.54.45	8.05	.56.25	1.40	60	34	27	30.5	27	27.75	.5			24.5	2,818	2,732	198
11.00.44	11.13.	46	11.10.30	9.	11.13.30	3.		40	23	31.5	25.5	28	.5			27	2,859	2,966	234
.15.55	.16.15	20	.24	7.45	.27.30	3.30		32	24	28	27	28	.5			23.5		3,155	189
.32.30	.33.15	45	.40.30	7.15	.43.35	3.05		42	24	33	26	28	.5				2,934	3,396	241
.47.50	.48.05	15	.54.40	6.35	.56.25	1.45		35	25	30	27	28	.5			24	2,975	3,579	183
12.01	12.01.45	45	12.11.35	9.50	12.14.20	2.45	15	33	28	30.5	26¼	28	.5			24	3,019	3,844	265
.17.30	.17.45	15	.24.34	6.49	.27.00	2.26		34	25	29.5	26	27	.5			24	3,043	4,023	179
.30. 5	.31.50		.40	8.10	.43.50	3.50		35	20	27.5	25½	28	.5				3,090		
.46.50	.48	70	.54	6.	.56	2.		26	22	24	27	28	.5			22.5	3,133	4,293	270
1.01.10	.02.10	60	1.10 25	8.15	1.14.15	3.50		30	20	25	26½	28						4,460	167
.18.30	.18.50	20	.25.55	7.05	.28.45	2.10		29	22	25.5	27	28						4,719	259
													.5			23	3,214		

Throttle was actually wide open, as further opening of valve produced no effect.

TABLE VII.—*Continued.*

	Full Speed.		Slow Down.		End.			Steam Pressure.			Vacuum.			Blow Off.					Counter.	
Time.	Time.	Interval.	Time.	Interval.	Time.	Interval.	Delay.	When Leaving.	On Arrival.	Average.	When Leaving.	On Arrival.	Position of Throttle Handle.	Beginning.	End.	Interval.	Revolutions per Minute.	Meter Readings.	Total.	Difference.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
H.M.S.	H. M. S.	M.S.	H.M.S.	M. S.	H.M.S.	M. S.	Scs.									Scs.				
1.32.13	1.33	47	1.41.35	8.35	1.44	2.25		33	24	28.5	26	28	0.5 Throttle was actually wide open.				26		4,502	
1.48.15	.48.32	17	.55.30	6.58	.58.55	3.25		33	24	28.5	27	27.5	"				23	3,274	4.750	248
2.02.20	2.03.10	50	2.11.45	8.35	2.14.40	2.55		32	24	28	26	27	"				26		4.930	180
.31.15	.31.30	15	.38.15	6.45	.41.45	3.30		35	24	29.5	25.5	27.5	"				25	3,365	5,168	238
.45.16	.47.10	1.54	.52.55	5.45	.59.50	6.55	70	35	25	30	25	28	"				26.5		5,361	193
3.03.10	3.03.20	10	3.10.15	6.55	3.12.55	2.40		37	28	32.5	27	28	"				25	3,449	5,618	257
.16.20	.17.10	50	.24.55	7.45	.28	3.05		35	29	32	26	28	"						5,801	183
.32.20	.32.30	10	.39.30	7.	Slip. .46.40	7.10		37	31	34	27	28	"				23	3,524	6,028	227
.49.30	.50.15	45	.57.45	7.30	4.00.13	2.28		33	24	28.5	25	27.5	"				25		6,254	226
4.04.50	4.05.05	15	4.12.40	7.35	.15.20	2.40		37	22	29.5	26.5	28	"				25	3,614	6,469	215
.19.05	.19.55	50	.27.10	7.15	.29.45	2.35	50	35	25	30	26.5	28	"				26	3,684	6,666	197
.32.35	.32.50	15	.40.45	7.55	.43.35	2.50		32	23	27.5	27	28	"				24		6,867	201
.47.20	.47.30	10	.54.40	7.10	.57.30	2.50		33	27	30	26	28	"				26	3,775	7,074	207
5.02.10	5.02.20	10	5.11.05	8.45	5.14.00	2.55	30	31	24	27.5	26	28	"	5.0159	5.0210	11	24.5	3,794	7,271	197
.17.30	.18	30	.24.55	6.55	.27.45	2.55	20	31	28	29.5	25	27	"				25		7,490	219
.31.50	.32.05	15	.40.30	8.25	.43.44	3.14		37	23	30	17.25		"				24	3,856	7,676	186
.47.43	.48	17	.54.35	6.35	.57.25	2.50		34		30	26.5	28					25.5		7,896	220
6.02.25	6.02.40	15	6.11.50	9.10	6.14.15	2.25		37.5	20.5	29	27	28					24	3,951	8,080	184
.22.20	.22.40	20	.30.15	7.35	.32.30	2.15		34.5	24	29.25	26.5	27.5							8,313	233
8.43.25	8.43.45	20	8.50.15	6.30	8.56.05	5.50		28		30	27							4,258	9,785	1,472
9.03.15	9.03.35	20	9.11.55	8.20	9.14.25	2.30		32	20	26	26	28					25		9,988	203
.23.25	.23.46	21	.32.55	9.09	.34.15	1.20	30	39	28	33.5	27	27.75					24.5	4,349	10,308	320
.42.10	.43	50	.51.15	8.15	.53.45	2.30		38	22	30	25.5	28		9.4150	9.4210	20	26		10,399	91
Total,		M. S. 25.50		H.M.S. 6.7.16		H.M.S. 2.20.0	M.S. 3.20			Average 32.62	Gauge, 2¼ lbs. too low.						Average, 25.799		10,631	232

From the table we deduce that during a total time of service amounting to 14 hrs., 39 mins., 40 secs.,

hrs.	mins.	secs.	
0	25	50	was devoted to "getting up" speed in leaving the dock.
6	7	16	" " running at full speed in crossing the river.
2	20	0	" " running slow with engine worked by "bar."
0	5	45	" consumed in delays in mid-river with engine at rest.
8	58	51	Total time boat was out of slips.

The throttle was practically wide open during the whole interval that the boat was away from the slips.

The speed during the interval of 25 mins. and 50 secs., devoted to attaining full speed, may be taken at half full speed. After slowing down, the average number of total revolutions made by the engine under the control of the "bar" was 12; the range being from 10 to 16. As there were 48 trips, we may therefore assume that $12 \times 48 = 576$ revolutions were made with the engine worked by the "bar." The total revolutions during the entire time were 10,631. The steam consumption per stroke when working by "bar" may be taken at twice that for regular action. Hence we may consider the total revolutions of the engine as equivalent to $10,631 + 576 = 11207$ revolutions at full throttle and regular valve action under an average boiler pressure of 32 lbs. above the atmosphere. The steam consumed was 2,091 cubic feet, each cubic foot representing 65 lbs. Hence, by weight the steam amounted to 135,905 lbs., which was evaporated in 14 hours and 40 minutes.

As the amount wasted by the safety-valve is insignificant, this weight of steam represents the consumption of the main engine and the steering-gear engines. The latter have two $5'' \times 5''$ cylinders, and made about 800 revolutions per hour during the test.

Allowing double the weight of steam displaced by the pistons and 7 per cent. clearance, the amount of steam consumed will be approximately 1 per cent. of that of the main engine.

To arrive at the average horse-power of the engine we may consider that 576 turns in 2 hours and 20 minutes, the time of "bar working," is equivalent to 3.2 revolutions per minute, or to a speed of about one-eighth of the average "full" speed of the engine. The mean effective pressure under the "bar" is about $\frac{5}{4}$ of that under cut-off. Hence, we may consider the action under "bar" as equivalent to $\frac{5}{4} \times \frac{1}{8} \times$ (2 hrs. and 20 mins.) at full speed, or to 28 minutes at full speed.

Again, the 25 mins. and 50 secs. devoted to "getting up" speed is equivalent to 12 mins. and 25 secs. at full speed. Hence we may conceive the total of 10,361 revolutions as made at full speed during 6 hrs. 7 mins. 16 secs. + 40 mins. 25 secs., or 6 hrs. 47 mins. 41 secs. This makes the average revolutions per minute 25.4.

The average mean effective pressure of the indicator cards, Fig. 84, was 31.9 lbs. Combining these two figures, we have as

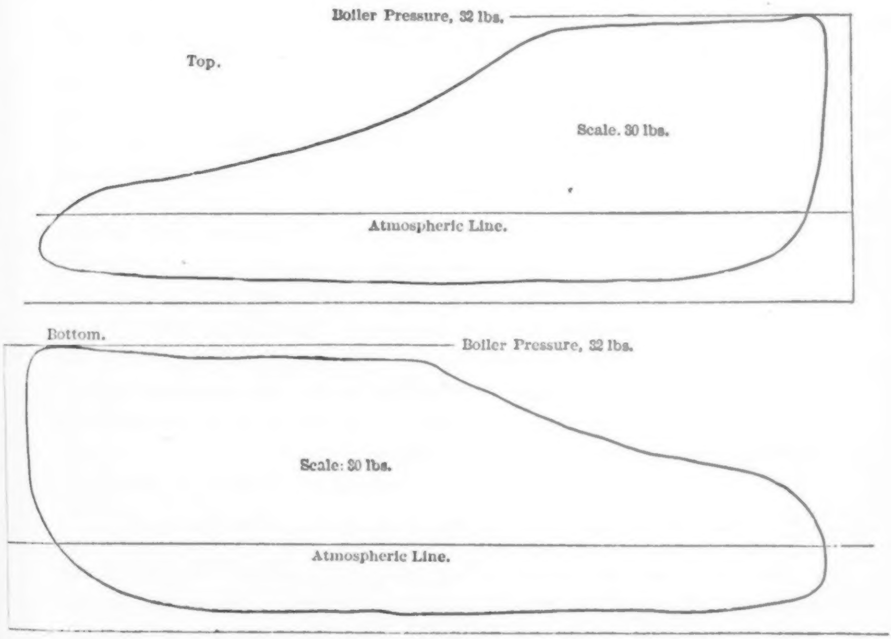


FIG. 84.
Average Card of the *Orange* in Ferry Service. Full Throttle.

the average horse-power of the engine in regular service 810 H.P.

Dividing this into the water consumption, 135,905 lbs., and the result by the time in which the engine developed the 810 H.P., viz., 6 hours 48 minutes, we have as the steam consumed per hour per indicated H.P., 24.7 lbs.

Subtracting one per cent. for the steering engines, the consumption of the main engines alone under 32 lbs. boiler pressure, $\frac{45}{100}$ cut-off, 25.4 revolutions, and 26 inches vacuum, becomes 24.45 lbs. per hour per H.P.

This figure must be taken as subject to an error of about one pound, in consequence of the nature of the conditions of the measurements, making the possible maximum 25.5 lbs.

Comparing with the consumption under the conditions of steady running, at 17 lbs. boiler pressure and 22.9 revolutions, which we have found to be 26 lbs. per hour per H.P. as a minimum, we find that the superior boiler pressure of 32 lbs., used in regular service, should make the steam per H.P. $\frac{82}{87}$* $= \frac{15}{16}$ of that for 17 lbs. boiler pressure. This gives for the probable theoretical consumption at 32 lbs. boiler pressure, $26 \times \frac{15}{16} =$ 24.45 lbs. The result is practically in agreement with the actual measurements, and it must, therefore, be concluded that the intermittent use of the engine in ferry service does not practically interfere with the realization of the maximum economy which would be expected of the engine under continuous service.

This conclusion should not be regarded as opposed by the fact that the engine cools by radiation during its periods of rest, and thereby induces extra condensation of steam during re-starting; for such cooling, during the short time a ferry-boat is at rest, lowers the temperature of the interior metallic surfaces so few degrees, that a single revolution of the engine may restore the metal to the average temperature at which it is maintained by the phenomena of cylinder condensation.

TEST OF LEAKAGE OF VALVES AND PISTON OF "ORANGE."

Inasmuch as the degree of possible tightness of poppet valves is subject to some variance of opinion, a careful test of the valves and piston was made as follows: The boat being in regular service, and the conditions of the engine, as regards expansion by heat, thoroughly normal, a hose leading to a condenser on scales was attached to the indicator cocks, and the sum of the leakage of steam through the valves and by the piston determined: 1st. When the pressure on both sides of the piston was

---

* This difference is due simply to the effect of a nearly constant back pressure in the two cases. The mean back pressure is 4.5 lbs. per sq. inch. This causes a loss of 18 per cent. of the mean forward pressure for 17 lbs. boiler pressure, but only 13 per cent. for 32 lbs. pressure. It follows that the steam per H.P. for 32 lbs. boiler pressure should be $\frac{1-0.18}{1-0.13} = \frac{82}{87}$ of that for 17 lbs. boiler pressure.

equal, both exhaust and admission valves being seated. 2d. When one side of the piston was connected with the condenser.

In each case the leakage was 48 lbs. per hour, thereby proving the piston tight, and that the entire leakage was through the valves. The leakage in service at ½ cut-off would be probably one-half of this amount, or ¼ per cent. of the total steam used.

The piston rings are expanded by their own elasticity with a force of about 5 lbs. per square inch. The engine had not been overhauled for about one year and had been steadily in use.

SPEED OF THE "ORANGE."

The boat left the foot of First Street, Hoboken, at 9.30 A.M., August 15, 1889, and arrived at Newburgh at 1.52 P.M. The distance by chart is 59 statute miles. The speed along shore was, therefore, $59 \div 4\frac{22}{60} = 13.50$ statute miles per hour. The time of high water was 11.16 A.M. at Governor's Island, and 2.45 P.M. at Newburgh. The tide was therefore at about the period of medium velocity during the whole run.

The medium velocity is known to be about 2.0 miles per hour, hence the speed through the water, or still-water speed, was 11.5 statute miles.

On the same run, the time to pass between posts 10 statute miles apart, near Yonkers, was 43 minutes 41 seconds, giving, for the speed along shore, 13.73 statute miles per hour, making the still-water speed 11.73 miles. A patent log gave for the same distance a still-water speed of 12.18 statute miles per hour, which would make the tide 1.55 miles per hour, but as shown in connection with tests of the *Bergen*, the log gives results about 1 mile per hour too great, which would make the tide 2.55 miles, and the corrected speed on log 11.18 miles per hour. Again, the log reading between Weehawken and Newburgh—a chart distance of 56 statute miles—was 48.38 statute miles between 10 A.M. and 1.52 P.M. This gives for the still-water speed 12.5 statute miles, which, corrected for error of log, becomes 11.5 statute miles.

On the return trip from Newburgh the times of high water and tides should have been approximately as follows:

	Time of High Water.	Time at which *Orange* Passed.	Tide, Statute Miles per Hour.
Governor's Island	11.16 A.M.		
Weehawken	11.46	6.20 P.M.	+ 3.0
Riverdale	12.46 P.M.	5.42	+ 3.0
Yonkers	1.00	5.30	+ 2.7
Irvington	1.16	5.00	+ 2.7
Haverstraw	1.30	4.15	+ 2.0
West Point	2.15	3.25	— 0.5
Newburgh	2.45	1.52	— 2.0

Multiplying tide values by the distance travelled by shore, we find the average equivalent to a constant tide of 1.25 statute miles with the boat.

The time to Weehawken was 4 hours 18 minutes, giving a speed with reference to shore of $56 \div 4.3 = 13.0$ statute miles. Hence the still-water speed was $13.0 - 1.25 = 11.75$ miles. The log gave a still-water corrected speed of 11.0 statute miles. On the same trip the boat passed between the 10-mile posts at Irvington and Riverdale in 40 minutes and 50 seconds, making the speed by shore 14.5 statute miles. The probable tide was 2.85 miles "with boat," making the still-water speed 11.65. The log gave a still-water corrected speed of 11.3 statute miles.

Summarizing these results, we have:

					Statute Miles per Hour.
Still-water speed	59	miles run,	Hoboken to Newburgh,	by Tide Table.	11.5
"	56	"	Weehawken "	by log	11.5
"	10	"	via " "	"	11.2
"	10	"	via " "	by Tide Table.	11.7
"	56	"	N'burgh to Weehawk'n,	"	11.75
"	56	"	" " "	by log	11.00
"	10	"	via Weehawken	"	11.3
"	10	"	via "	by Tide Table.	11.7
Average					11.5

The indicated horse-power and revolutions averaged 538 during the trip to Newburgh and 560 during the return—the revolutions varying from 23.4 to 22.5.

The average horse-power both ways is 490 and the revolutions 22.9 per minute. These figures and the above average speed should be considered liable to an error of at least 5 per cent.

THEORETICAL SPEED OF "ORANGE" FOR 490 H.P.

The wetted surface and obliquity of water lines of the *Orange* are as follows:

	Immersed Surface. Square Feet.	Maximum Angle of Water Lines.	Co-eff. of Aug. $1 + 4 \sin^2 + \sin^4$.
Keel and rudders..........	500	0°	1.000
Keel to 2 ft. line............	1211	17° 45′	1.469
2 ft. line to 4 ft. line........	1604	15° 15′	1.368
4 ft. line to 6 ft. line...... ..	1456	17° 30′	1.280
6 ft. line to 7 ft. 4 in. line....	800	19° 45′	1.380
Total...................	5571	Av. 14° 0	Av. 1.319

Multiplying each co-efficient by the surface controlled, the average value of the co-efficient of augmentation is 1.319.

The augmented surface is therefore

$$5{,}571 \times 1.319 = 7{,}347 \text{ square feet.}$$

By the formulæ of Rankine's shipbuilding treatise the force, in pounds, to propel a hull through the water, is

$$\text{Augmented surface} \times \frac{(\text{speed in knots per hour})^2}{100}.$$

Hence, the force to propel the *Orange* at one knot per hour should be 73.47 lbs. One knot per hour is 100.8 feet per minute. Hence, to propel the *Orange* at one knot per hour should require

$$\frac{73.47 \times 100.8}{33{,}000} = 0.225 \text{ H.P.}$$

Assume that 25* per cent. of the indicated horse-power is wasted in the frictional resistances of the engine and disturbances in the motion of the water other than slip. Assume also that the slip is 27* per cent. Then only $0.75 \times 0.73 = 0.55$ of the indicated horse-power is available for the propulsion of the hull.

This fraction of 490 H.P. is 269 H.P. Hence

$$\sqrt[3]{\frac{269}{0.225}} = \sqrt[3]{1{,}210} = 10.65,$$

is the speed in knots per hour which should result from 490 in-

* The feathering-paddle steamer *Admiral* was of nearly the same size and power as the *Orange*, with the same area of paddles and paddle wheels. Rankine found, from his own experiments with this vessel, a frictional loss of 23 per cent., and a slip of 22 per cent. It is therefore reasonable to increase these figures for the radial paddles of the *Orange*. This allowance for friction is also consistent with the calculated friction of the machinery with a co-efficient of friction of 10 per cent.

dicated horse-power. This calculation of speed amounts to using a constant of 18,000 instead of 20,000 in Rankine's formula,

$$H.P. = \frac{\text{speed}^3 \times 20{,}000}{\text{augmented surface}}.$$

The result is equivalent to a still-water speed of 12.25 statute miles, a figure not beyond the probable limit of error of the observed speed of the boat.

THEORETICAL SPEED CALCULATED FROM SLIP OF PADDLES OF "ORANGE."

The paddle-wheels of the *Orange* are 21 feet diameter to outside of floats. The latter are 20 inches deep (radially) and 4 feet long. They are distributed over 20 spokes on each half of the wheel, a float on the outer half being midway between two floats on the inner half. When four floats are submerged on one-half of the face three are submerged on the other half. When the boat is at rest the outer edge of a vertical float is about 33 inches from the surface of the water. If the centre of pressure is assumed to be 11 inches from the outer edge of the floats, then the slip for 22.9 revolutions per minute is 27 per cent. for a still-water speed of 11.5 statute miles per hour, and 23 per cent. for a speed of 12 miles per hour. Let it be assumed that the stream of water acted upon by the paddles is equal to the maximum submergence of 33 inches times the total face of the wheel, or 8 feet; then the total area of stream reacted by both paddles together is 44 square feet. Then, if the wheels send astern each second a volume of water equal to this area times the velocity of the centre of pressure of the wheels per second = actual speed of boat ÷ (1 − slip). The velocity which this water receives is equal to the slip expressed in feet per second:

$$= \frac{\text{actual speed of boat}}{1 - \text{slip}} \times \text{slip}.$$

The force required to be exerted by the paddles to create this velocity is the product of the latter by the weight of the volume of water sent astern, divided by gravity, or 32.

Hence the force exerted by the paddle is

$$\frac{(\text{speed of boat})^2}{(1 - \text{slip})^2} \times \text{slip} \times 48 \times \frac{63}{32} \text{ lbs.};$$

in which 63 is the density of the water. If this force be multiplied by the feet moved per second by the boat in still water

the product is foot-pounds of work, and if divided by 550 the quotient should be the horse-power to propel the hull, and the latter should be 0.55 of the indicated horse-power, as previously explained.

Hence,

$$0.55 \times 490 = \frac{(\text{speed of boat})^3}{(1 - \text{slip})^2} \times \text{slip} \times \frac{44 \times 63}{550 \times 32},$$

in which the speed is expressed in feet per second.

Hence, speed in statute miles per hour should be

$$\frac{3600}{5280}\sqrt[3]{\frac{0.55 \times 490 \times (1 - \text{slip})^2 \times 32 \times 550}{44 \times \text{slip} \times 63}}.$$

For 27 per cent. slip, this gives a speed of 10.23 miles.

For 23 per cent. slip, this gives a speed of 11.25 miles.

Either a less value for slip or a less value for the area of the stream acted upon is necessary to make these figures agree with the results of observations of speed. For example, if the slip was 20 per cent., the speed estimated from the slip, with the same area of stream, would be 12 miles. A reduction of slip to this figure would be caused by either a change of speed of revolutions to 22.5 per minute or by the assumption of a speed of 12.5 miles an hour. Neither of these alterations of data are, however, warranted within the probable limits of error. On the other hand, with a reduction of the area of stream equal to 20 per cent., a slip of 23 per cent. corresponds to a speed of 12 miles, which agrees with the upper limit of observed speed. Such a reduction of area involves a reduction of 6 inches of dip of paddle, and it is not unreasonable to suppose that some approximation to this may be caused by the shape of wave-surface produced at the paddle-wheel by the motion of the boat. Besides these possibilities for modifying the data, the distinction between real and apparent slip should be considered, as explained in the discussion of the subject by Rankine.

It is probable that the difference between the real and apparent slip is slight, but whatever difference may exist would assist in increasing the speed calculated from slip.

A most important conclusion to be drawn from this whole discussion of the speed of the *Orange* is that determinations of speed based upon long runs in the midst of tidal currents should

not be used for scientific deductions. As will be seen in connection with the *Bergen*, the discrepancies of speed determinations on even a short course, due to tides, are scarcely controllable without the most elaborate arrangements for eliminating the tide.

SPEED OF "ORANGE" IN FERRY SERVICE.

By Column 5 of Table VII., the average time at full engine speed between Hoboken and Barclay Street, New York City, during 14 hours, was $7\frac{1}{2}$ minutes. The distance, allowing for the curved course generally followed during the half-hourly night trips, is $1\frac{3}{4}$ statute miles. After the engine is at full speed, some time elapses before the boat is under full headway. This time seems to be about 30 seconds, during which the boat goes about 300 feet. About 300 feet must also be allowed for the distance the boat is away from the slips when the engine is slowed. These allowances make the distance run at full speed 1.65 miles in 7 minutes. This is equivalent to 14.0 statute miles per hour.

The indicated horse-power during the service test was found to be 810. If the boat were driven 12 miles per hour, with 490 H.P., the speed for 810 H.P. should be

$$12 \times \sqrt{\frac{810}{490}} = 14.18 \text{ statute miles.}$$

Within the limits of error of these figures, it may be concluded that the boat is able to travel across the river at the rate of 12 miles, with reference to the shore at all stages of the tide.

PART II.

EXPERIMENTS WITH SCREW-BOAT "BERGEN."

STEAM ECONOMY OF "BERGEN."

Description of Machinery.

The boilers are shown in Fig. 86. They are of the locomotive type in that the products of combustion traverse the length of the boiler but once before arriving at the chimney. The boilers

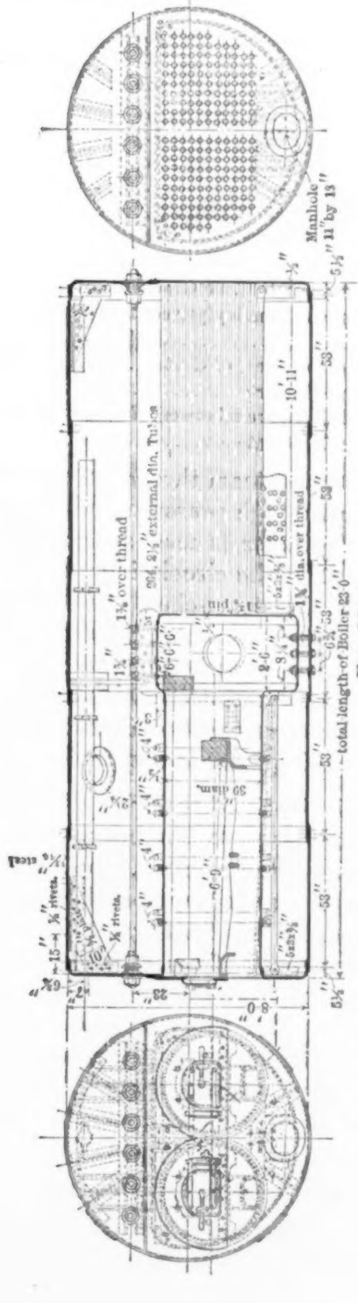


FIG. 86.

Boiler of the *Bergen*.—Two boilers are used with a connecting steam drum, 32 inches diameter, 19 feet long.

are two in number, and the principal dimensions of each boiler are as follows:

Diameter of Shell	8 ft.
Length	23 "
Thickness of Shell	$\frac{1}{2}$ in.
Diameter of Furnaces	$3\frac{7}{8}$ ft.
Thickness of "	$\frac{5}{8}$ in.
Heating Surface in Furnaces	192 sq. ft.
" " " Combustion Chamber	82 "
" " " in Tubes, 204—$2\frac{1}{2}''$ diameter	1,457 "
Superheating Surface	0 "
Cross Section of Tubes per cent. of Grate	18
One-half Cross Section of Chimney per cent. of Grate	18
Grate Surface	40 sq. ft.
Opening in Grate Bars per cent. of Grate	50
Ratio Heating to Grate Surface	43
Steam Space—one-third in a drum between the two boilers	300 cubic ft.
Height of Chimney above Grate	53 ft.

The main engines take steam through a separator attached to the steam-pipe near the high pressure steam-chest. The jacket space around each cylinder is used as the steam-chest for that cylinder. Steam to drive the circulating pump is taken from the first receiver, that is, the steam-chest of the intermediate cylinder. Steam to drive the feed pump, bilge pump, ventilating and steering engines, is taken directly from the boiler through reducing pressure valves set to maintain 30 lbs. pressure.

The pumps, etc., are arranged to exhaust either into the main condenser or overboard against atmospheric pressure.

SEPARATION OF STEAM CONSUMPTION OF MAIN ENGINES.

During the 120-mile run to Newburgh the pumps, steering engines, etc., were made to exhaust overboard, and only the condensed steam from the main engines allowed to collect in the hot well. Consequently the total water fed to the boilers was greater than the condensed steam entering the hot well. The deficiency was supplied from the large water tanks, which were kept filled to a mark. The water supplied to these tanks was sent through one water meter, and the total feed water of the boilers was sent through a second meter. The difference of the amounts shown by the two meters was the consumption of the main engines alone.

BOILER TEST, 120-MILE RUN.

On the day previous to that of the run to Newburgh the heating surfaces of the boilers were swept free of soot and steam

gotten up. The fires were then banked until 4 A. M. the next day. At this time the banks were spread, steam raised to 140 lbs., and the engines run a short time, up to 5 A. M., when the fires were drawn and a new wood fire kindled at 5.35 A. M., with 717 lbs. of wood. The steam pressure fell to 110 at 5.40, and commenced to rise again at 6.00. At 7.00 steam was 140 lbs., and the dampers were partly closed to hold the same pressure until 9 A. M., the engine being worked very slowly until this time. At 9 A. M. the boat steamed up the river, and coal was burned until 6.30 P. M., at the rate of about 1,600 lbs. per hour. The boat arrived at her dock at 7.20 P. M., and the fires were allowed to burn out until 8.10 P. M., when steam had fallen to the same pressure at which the test was started, 110 lbs.

The total coal consumed, including the kindling wood, taken as equal to $\frac{4}{10}$ as much coal, was 15,906 lbs., making 1,116 lbs. per hour for the whole interval from 5.35 A. M. to 8.10 P. M. The total water evaporated was 146,548 lbs., making the evaporation, per lb. of coal, 9.21 lbs. of steam at 114 lbs. pressure and 118° temperature of feed water.

The ashes amounted to 7.9 per cent. of the coal, making the evaporation, from and at 212°, 11.4 lbs. of steam per lb. of combustible.

The rate of combustion was 20 lbs. of coal per sq. foot of grate per hour during the interval from 9 A. M. to 6.30 P. M., but much less before and after this period. A twenty-six hour test in ferry service gave an economy of 8.4 lbs. of steam per lb. of coal, for a rate of combustion of $10\frac{1}{2}$ lbs. per hour per sq. foot of grate—a result which is possibly explainable on the supposition that the ashes were a greater per cent. of the coal than during the Newburgh trip. A difference in the per cent. of ashes of 3 per cent. in 10 per cent. is sufficient to nearly account for the difference between the evaporation of 8.65 lbs. of water in the case of the *Orange*, and the 9.2 found for the *Bergen's* boiler on the Newburgh trip.

The steam consumption and H.P. during 120-mile run, from 9.30 A. M. to 6.30 P. M., is shown by Table VIII.

TABLE VIII.

FERRY-BOAT "BERGEN,"

120-MILE TEST OF STEAM CONSUMPTION, SEPTEMBER 18, 1889.

Time.	Boiler Pressure. Gauge lbs. per sq. in.	Revolutions per minute.	Total horse power.	REMARKS.
9.30	110	144.4	741.9	
10.	112	144.4	744.4	
10.30	114	145.	708.0	
11.	110	145.1	696.0	
11.30	108	142.1	684.7	Throttle wide open, as per Fig. 79.
12.	110	142.1	677.0	Average Horse-power, 663.
12.30	105	142.1	657.7	
1.	95	139.9	613.3	
1.30	90	132.9	532.5	
2.	95	133.3	570.6	
2.30	140	....	590	At Newburgh.
3.	130	139.7	625	
3.30	125	144.3	654	
4.	120	142.2	654	Steam throttled as per cards *a* and *b*,
4.30	$122\frac{1}{2}$	142.8	654	(Fig. 78).
5.	120	143.	654	Average Horse-power, 670.
5.30	122	145.	733	
6.	119	145.	733	
6.30	120	145.	733	

The average indicator card while the throttle was wide open is shown in Fig. 79. For low card see page 434.

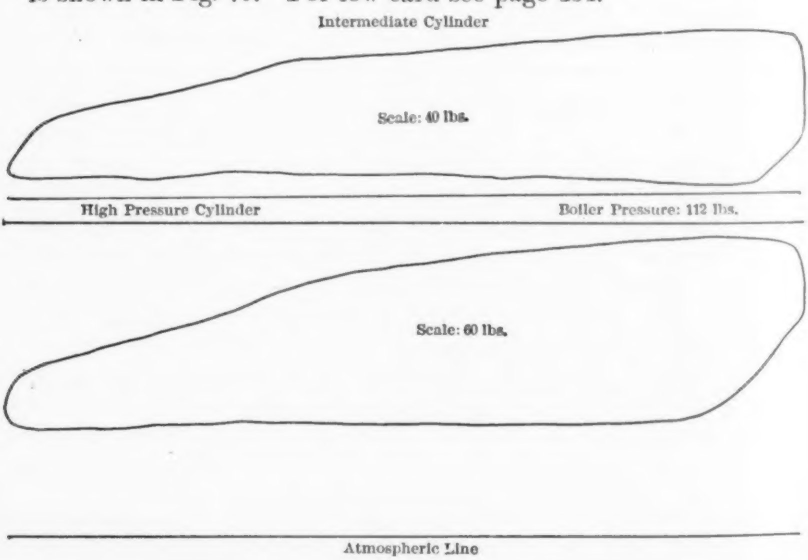


FIG. 79.

Average cards, throttle wide open, 120-mile run.

The extent of the throttling during the return trip is shown by the high cylinder card, Fig. 78.

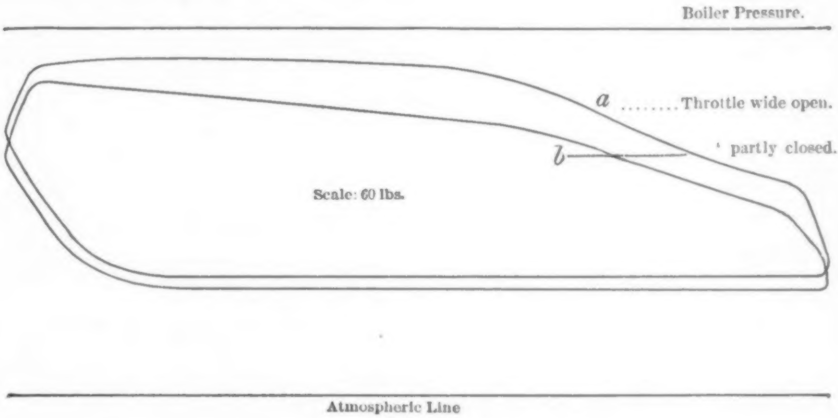


FIG. 78.

The average indicated horse-power of the main engines is placed at 665.

The total steam consumed was 14,411 lbs. per hour, of which 2,358 lbs. was consumed by the feed, circulating, and bilge pumps, ventilating, and steering engines.

The consumption per hour per H.P. was therefore 21.7 lbs. for all purposes, and 18.3 for the main engines alone.

As checks upon the direct determination of the steam used, the following estimate is made:

ACCOUNT OF HEAT RECEIVED AND REJECTED.

By the measurement of water fed to the boiler, it was determined that the steam used by the main engines per hour was 12,153 lbs. at 115 lbs. pressure. This steam contained not more than one per cent. of moisture by the calorimeter test. Hence its total heat above zero was $1219.3 - 8.69 = 1210.6$ British thermal units, by Regnault's determinations.

When this steam enters the condenser, there has already been taken from it by the engine-pistons 660 horse-power of work, or

$$\frac{660 \times 1{,}980{,}000}{772} = 1{,}692{,}746 \text{ British thermal units, or}$$

$$\frac{1{,}692{,}746}{12{,}153} = 140 \text{ units per pound of steam, and also a cer-}$$

tain amount due to the radiation of the cylinders. There is about 150 sq. feet of radiating surface, which may lose, as a maximum, 0.75 heat units per hour per square foot per degree difference of temperature between the steam and the atmosphere. Taking this difference as 275°, we have the loss per hour

$$0.75 \times 150 \times 275 = 31{,}000 \text{ heat units, or}$$

$$\frac{31{,}000}{12{,}153} = 2.5 \text{ heat units per pound of steam.}$$

Hence, on entering the condenser each pound of steam contains

$$1{,}210.6 - 140 - 2.5 = 1{,}068.1 \text{ thermal units.}$$

The total heat delivered to the condenser per hour is, therefore,

$12{,}153 \times 1068.1 = 12{,}980{,}619$ thermal units, and this amount of heat should be found in the sum of the three* following quantities :

1st. Heat removed by the circulating water.

2d. Heat absorbed by water drawn from fresh-water tanks and mixed with condensed steam in the condenser.

3d. Heat remaining in condensed steam when it arrives in the hot well.

The amount of circulating water, taking it to be half of the time salt, and half of the time fresh, so that its weight per cubic foot is 63 lbs., is :

Area of plunger.		Observed stroke.				Single strokes per hour.		Density.	
1.07	×	1.03	×	4	×	1339	×	63	= 371,879.

The specific heat of salt water or brine of the density of sea water is 0.96 for 60°. Assuming it to be the same at 104°, we have the average specific heat of the circulating water, taken as half of the time fresh,

$$\frac{0.96 + 1.0}{2} = 0.98.$$

The range of temperature through which the circulating water was heated in passing through the condenser was 31.4°. Hence the heat removed by the circulating water was

$$371{,}879 \times 0.98 \times 31.4 = 11{,}443{,}460 \text{ thermal units.}$$

* Radiation from the condenser and hot well is neglected, as its amount is unimportant.

If the slip of the pump be assumed at 2 per cent., this figure becomes 11,214,591 thermal units.

The water drawn from the tanks was 2,358 lbs. per hour, and was at 75°. The average temperature in the hot well was 118°. Hence this water absorbed

$$2{,}358 \times (118 - 75) = 101{,}394 \text{ thermal units.}$$

The heat remaining in the condensed steam was

$$12{,}153 \times 118 = 1{,}434{,}054.$$

The sum of these three quantities gives:

Absorbed by circulating water	11,214,591
" " tank "	101,394
Remaining in steam	1,434,054
Total heat rejected	12,750,039
" " received	12,980,619
Discrepancy	230,580

This discrepancy represents less than $\frac{1}{4}$ inch variation of pump stroke, and as the length of the latter was scarcely measured within this limit of error it may be considered that the measurement of heat rejected satisfactorily confirms the measurement of steam used, as determined from the feed-water.

QUALITY OF THE STEAM.

A Barrus Superheating Calorimeter was attached to the main steam-pipe near the high steam-chest. It showed the steam to contain $\frac{3}{4}$ of one per cent. of moisture. The separator on the steam-pipe showed no water by its water-glass until about 1 P. M., when about 50 lbs. of water was present. This was blown out and about the same amount had again collected at the end of the test.

THEORETICAL STEAM CONSUMPTION OF "BERGEN."

The combined action of the steam in the three cylinders of the *Bergen* is shown in Fig. 81.

The action is equivalent to that of a single cylinder equal to the low cylinder of the *Bergen*, in which steam at 100 lbs. pressure should be cut off at about $\frac{1}{22}$ of the stroke, with a clearance of 11.3 per cent., with the following modifications:

I. There is a loss of mean effective pressure, represented by

the areas (a) and (b), due to the fact that the the pistons are not in motion while the steam expands into the clearance spaces of the intermediate and low cylinders respectively.

II. There is a gain of mean effective pressure, represented by the shaded areas (c) and (d), due to the fact that the clearance volumes in the triple cylinder engine are less than in the single engine, by amounts represented by these areas; and hence,

III. If the sum of areas (a) and (b) is less than sum of (c) and (d), as is here the case, the net result is that the mean effective

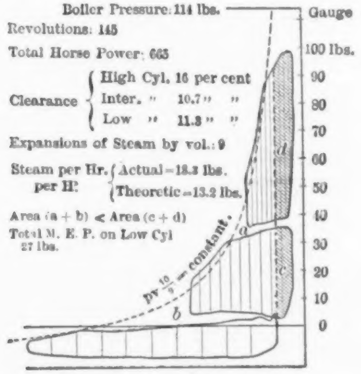


FIG. 81.

Average Card of *Bergen*, 120-Mile Run. Throttle wide open. H.P., 665.

pressure is nearly that due to expansion in a single cylinder with a loss due to a per cent. of clearance equal to that of the single engine *divided* by the ratio of the volume of the low to the high cylinder, which is about 5 to 1.

IV. Consequently the theoretical consumption per hour per H.P. corresponds to a real ratio of expansion of about 9 and a clearance of about $2\frac{1}{2}$ per cent in a single cylinder, whereas the apparent expansion estimated from the cut-off in the high cylinder is about $7\frac{1}{2}$ times, with 16 per cent clearance.

ECONOMY OF THE ENGINES OF THE "BERGEN" COMPARED WITH THOSE OF SS. "METEOR." *

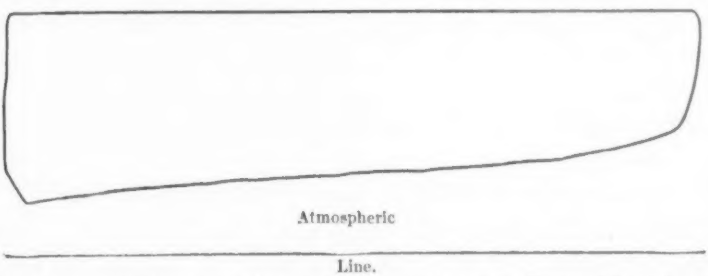
The only record of water consumption of a triple expansion marine engine under the pressure for which the *Bergen's* engine was designed is that of the SS. *Meteor*. In this case the

* The engines of this ship were accurately tested by the Committee on SS. Economy of British Association of Mechanical Engineers.

engines were of about 2,000 H.P. capacity. The clearances were nearly the same as for the *Bergen*. The boiler pressure was about 150 lbs. and all of the cylinders were jacketed with steam of pressure about equal to the admission pressure. The difference of the boiler pressure and greater range of expansion in the case of the *Meteor* indicates that her engines should consume less steam per hour per H.P. than the engines of the *Bergen* in the ratio of 11.2 to 13.2, which would make the *Meteor's* consumption $\frac{11.2}{13.2} \times 18.3 = 15.5$ lbs. The *Meteor's* actual consumption was 14.9 lbs. The difference is about 3 per cent., which may be attributed either to the jackets of the *Meteor* or to the errors of determination in the case of the *Bergen*, the measurement in the case of the *Meteor* having been too carefully made to regard it as in error to the above extent.

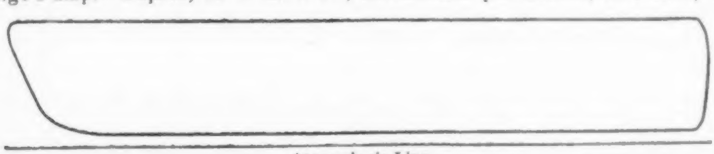
HORSE-POWER OF CIRCULATING PUMP.

This pump is of the direct-acting, duplex type, with steam cylinders, 12 inches diameter, and water-plunger, 14 inches di-



Feed Pump.—Duplex, 12″ × 12″ × 4½″ ; Revolutions per Minute, 55 ; H.P., 9.

Bilge Pump.—Duplex, 12″ × 12″ × 5″ ; Revolutions per Minute, 48 ; H.P., 1½.



Circulating Pump.—Duplex, 12″ × 18″ × 14″; Revolutions per Min., 22 ; H.P., 2.7

FIG. 82.

Average Pump-Cards of the *Bergen*. Scale, 20 lbs. to the Inch.

ameter. The possible stroke is 13 inches, but its average piston-travel during the test was 12¼ inches. It ran steadily and made 1,339 revolutions per hour. Indicator cards from its steam cylinder averaged as per Fig. 82. The mean effective pressure averages 8.5 lbs. per square inch. Its indicated power is, therefore,

$$\frac{114 \times 8.5 \times 4 \times 1.02 \times 22.3}{33,000} = 2.7 \text{ H.P.}$$

The amount of water pumped was 5,844 cubic feet per hour, which was lifted through 4.5 feet. The work corresponding to this lift is 0.84 H.P., so that 1.86 H.P. is consumed in overcoming the resistance to the passage of the water through the condenser, and in the friction of the pump mechanism. The friction of such pumps is known to be about 3 per cent. of the indicated power.

Hence, the horse-power to circulate the water through the condenser was 0.31 H.P. per 1,000 cubic feet of water circulated which corresponds approximately to 100 H.P. of work by the main engines.

The steam consumption estimated from the card, Fig. 82, is	396	lbs. per hour.
Probable amount due liquefaction*	170	" "
Total per hour	566	"
Probable consumption per hour per H.P., non-condensing	210	"
" " " " condensing	130	"

HORSE-POWER AND STEAM CONSUMPTION OF FEED-PUMP.

The boiler feed pump is a direct-acting duplex, with 12″ x 12″ steam cylinders, and 4½″ water cylinders. In drawing its water from the hot well, considerable air is absorbed, owing to the variable condition of the level in the hot well. To eliminate such air, a pipe is led from the base of the air-chamber of the pump to the top of the hot well, and a large portion of the pump's dis-

* The steam condensed by the metallic surfaces is estimated, in the case of all of the pumps, by the following formula:

$$\text{Pounds condensation per stroke} = \frac{\text{Surface exposed during admission} \times \text{time in seconds} \times \text{range of temperature}}{60}$$

The constant, $\frac{1}{60}$, is deduced from experimental determinations of consumption of an 8″ × 10″ pump. The total estimated consumption of the pumps and ventilating engine is 2,400 lbs. The amount given by the measurements was 2,358 lbs.

placement simply circulates between the latter and the hot well. The speed of the pump, therefore, amounts to about four times the number necessary to displace the volume of feed-water, the revolutions per hour averaging 3,250 per hour during the trip to Newburgh and return.

The pump is supplied with steam through a "reducing pressure valve," which maintains 30 lbs. pressure in the steam-chest of pump. The mean effective pressure was 15 lbs., as per Fig. 82, making the indicated work equal to 9 horse-power.

The steam consumed per hour, estimated from the card, is...	1,170 lbs.
Probable amount consumed by cylinder condensation.......	250 "
Total...........	1,420 lbs.
" per hour per H.P..........	158 "

If the pump exhausted into the condenser, as is the case in regular service, the probable consumption by card would be 25 per cent. less, making the amount per H.P. 125 lbs.

BILGE PUMP AND VENTILATING AND STEERING ENGINES.

The bilge pump is 12"x 12"x 5", and made 2,930 revolutions per hour during the Newburgh trip. Its indicated horse-power, Fig. 82, was 1.5, and the estimated steam consumption 575 lbs. of steam per hour, 10 per cent. of which is estimated as due to liquefaction. The steam per hour per H.P. is, therefore, about 380 lbs. By use of the vacuum the consumption of this pump would be about one-third as much, or 130 lbs. per hour per horse-power.

The ventilating engine ran constantly at about 300 revolutions per minute during the Newburgh trip. This engine is 3 inches by 6 inches, and is estimated to have consumed 117 lbs. of steam per hour during the Newburgh trip.

The steering engines used very little steam during the trip to Newburgh. The whole amount would be a fraction of 1 per cent. of the entire consumption.

ECONOMY OF SURFACE CONDENSATION AND OF USE OF DIRECT-ACTING PUMPS.

By the use of a surface condenser the cost of water for the steam used by the engines is almost entirely avoided, but the coal used by the circulating pump is an extra expense as compared to using a jet condenser. The amount of fresh water saved is, as a maximum, 225 cubic feet per hour. This represents about

25 cents per hour of cost. The coal consumption of the circulating pump, exhausting into the condenser, assuming $8\frac{1}{2}$ lbs. evaporation, is about 50 lbs. per hour, representing 10 cents of cost. If a centrifugal pump was used as a circulator, the steam consumption of its engine might be 45 lbs. per hour per H.P. instead of 130, as in the case of the direct-acting pump; but the power required would be about 2 times as great, as the efficiency of the centrifugal would be only about half of that of the steam pump. The net saving, therefore, would be $130 - 2 \times 45 = 40$ lbs. of steam per hour per horse-power, or, say, a total of 110 lbs. of steam per hour. This represents about 3 cents per hour of possible saving by a centrifugal circulator. A greater economy could be obtained by a crank and fly-wheel pump, but the simplicity of the direct-acting pump makes it on the whole probably the best device. If the exhaust steam of the pumps, etc., was made to heat the feed-water from the hot well, so that the latter would enter the boiler at 212° Fahr., the result would be as follows: The pumps, etc., would exhaust against the atmosphere, and their combined consumption would be 2,358 lbs. per hour, which is about one-sixth of the total feed-water. The latter would leave the hot well at 118°. To heat it to 212° would require 95°. As the water to be heated would be six times the weight of the steam supplying the heat, each pound of the steam must supply six times 95°, or 570°. As it would be necessary to abstract 966° in order to reduce all of the pump steam to liquid, it follows that only about $\frac{6}{10}$ of the pump steam could be saved. This would amount to about 2 lbs. of steam per H.P. of the main engines. The most economical arrangement would follow if the steam remaining vaporous after passing through the heater could be sent into the second receiver and thence used in the low-pressure cylinder.

AIR SUPPLY PER POUND OF COAL, AND AMOUNT OF HEAT WASTED BY CHIMNEY.

The velocity of the air flowing into the ash pits was measured by a Casella Anemometer, and found to average 461 feet per minute, as per Column 1 of Table IX. The joint area of the ash-pit openings was 8.9 square feet. The total amount of air supplied was, therefore,

$$8.9 \times 461 \times 60 = 246{,}174 \text{ cubic feet per hour.}$$

The coal burnt per hour between 9.30 A.M. and 6.30 P.M. averaged 1,600 pounds.

Hence, the air per pound of coal was 162 cubic feet, or, taking the density of air at 1 lb. to 12.5 cubic feet, we have the pounds of air supplied per pound of coal equal to 13.

By analysis of the chimney gases, the average composition of the latter was by Columns 2, 3, and 4, Table IX :

Carbonic acid	12.3	per cent.
Oxygen	7.0	" "
Carbonic oxide	1.1	" "

measured by volume. This requires that each pound of carbon should have supplied to it as many pounds of air as are represented by the following formula:

$$\text{Air per pound of carbon} = 4.3\left[\frac{4}{3}.\frac{2\,(CO_2 + O) + CO}{CO_2 + CO.}\right],$$

the symbols CO_2, O, CO, representing percentages by volume, as given by the analysis.

Hence, in the present case, we have :

$$\text{Air per pound of carbon} = 4.3\left[\frac{4}{3}\times\frac{2\,(12.3 + 7) + 1.1}{12.3 + 1.1}\right] = 17.0.$$

Assuming the combustible to represent practically pure carbon, and the combustible being 92.2 per cent. of the coal, we have the air per pound of coal by analysis of the gases equal to $17 \times 0.922 = 15.6$.

The bituminous coal may be assumed to have a composition as follows :

Ashes, clinker, and unburnt coal	7.8	per cent.
Moisture	3	" "
Hydrogen	6	" "
Carbon, volatilized to CO_2 or CO	80	" "
Oxygen, etc.	4	" "

The air required for combustion, assuming the carbon to all form CO_2, is

$$11.6 \times C + 34.8\,(H - \tfrac{O}{8}),$$

which gives in round numbers 11.0 pounds of air per pound of coal. The above figures, therefore, provide an excess of air, although they are lower than is usual with natural draught.

Taking the air per pound of coal at the average of the determinations by anemometer and analysis, we have 14.3 pounds.

Adding the combustible, the total weight of gas passing up chimney per pound of coal is

$$14.3 + 0.922 = 15.42 \text{ lbs.}$$

The specific heat of such gas is 0.24.

The average temperature of the gas at the entrance to the chimney was 750°. The temperature of the atmosphere was 80°. Hence the heat escaping in the chimney gases per pound of coal was

$$(750 - 80) \times 15.32 \times 0.24 = 2{,}528 \text{ thermal units.}$$

TABLE IX.

PRODUCTS OF COMBUSTION OF FURNACE, 120-MILE TEST OF "BERGEN."

Time.	Velocity of air entering ash pit, ft. per minute.	Composition of chimney gas. Percentages by volume.				Temperature of Chimney.		Draught gauge at top of uptake, inches of water.
		CO^2	O	CO	IV	Top.	Bottom.	
A. M.	(1.)	(2.)	(3.)	(4.)	(5.)	Several observations indicated the average to be 650° Fahr.	Not taken regularly, but several observations indicated the average to be 750° Fahr.	
10 00	511	11.0	8.4	0.	80.6			¼"
10.30	547	11.6	7.8	0.	80.6			"
11.30	509	12.8	6.0	0.	81.2			"
12.30	490	11.5	7.5	0.6	80.4			"
P. M.								
1.30	445	12.6	5.4	0.4	81.6			"
2.30	455	12.6	4.6	6.4	82.4			$\frac{5}{16}$"
4.30	443	13.0	5.6	1.0	80.4			"
5.30	411	14.2	3.2	2.4	80.2			"
6.30	395	12.4	5.6	1.0	81.0			"
Av.	461	12.3	7.0	1.1	81.0	650°	750°	$\frac{9}{32}$"

DISTRIBUTION OF HEAT OF FUEL.

The probable composition of the coal, as already given, would afford the following amount of heat by combustion:

Hydrogen	$\left(0.06 - \frac{O}{8}\right) \times 62{,}500 =$	3,438	thermal units.	
Carbon...........	$0.80 \times 14{,}500 =$	11,600	"	"
		15,038	"	"

Less heat to evaporate moisture and superheat it to temperature of chimney, or

$$0.03(750-212) \times 0.48 + 0.03(966+212-80) = \quad 41 \quad \text{“} \quad \text{“}$$

If some of the carbon burns to CO, according to the following analysis of flue gas,

$$
\begin{array}{llll}
CO_2 - 12.3 & \text{per cent.} & \text{by} & \text{volume.} \\
O \;\; - 7. & " & " & " \\
CO - 1.1 & " & " & " \\
N \;\; - 79.6 & " & " & "
\end{array}
$$

Then the percentages by weight are

$$CO_2 = \frac{12.3 \times 22 \times 100}{(12.3 \times 22) + (7 \times 16) + (1.1 \times 14) + (79.6 \times 79)} = 17.9 \text{ per cent.}$$

$$CO = \frac{1.1 \times 14 \times 100}{(12.3 \times 22) + (7 \times 16) + (1.1 \times 14) + (79.6 \times 1.4)} = 1.0 \;\; " \;\; "$$

The total carbon is

$$\left(\frac{12}{44} \times 17.9\right) + \left(\frac{12}{28} \times 1.0\right) = 4.88 + 0.43 = 5.31 \text{ per cent.,}$$

of which $\frac{0.43}{5.31} = 8.1$ per cent. burns to carbonic oxide, yielding 4,400 thermal units per pound, and 91.9 per cent. burns to carbonic acid, yielding 14,500 thermal units per pound. Out of the 0.8 pound of carbon in a pound of coal,

$$
\begin{array}{l}
0.081 \times 0.8 = 0.065 \text{ lbs. produce } 0.065 \times 4,400 = 286 \text{ thermal units;} \\
0.919 \times 0.8 = 0.735 \;\; " \;\; " \;\; 0.735 \times 14,500 = 10,657 \;\; " \;\; "
\end{array}
$$

Hence, instead of realizing from the 0.8 lb. of carbon 11,600 units by complete combustion to carbonic acid, we have only

$$286 + 10,657 = 10,943 \text{ units.}$$

Hence, the presence of 1.1 per cent. by volume of carbonic oxide in the chimney gases causes a loss of 657 units of heat per pound of coal.

The effect of moisture and hydrogen would be the same as before, making the net heat evolved by the combustion of the coal

$$10,943 + 3,438 - 41 = 14,340 \text{ thermal units.}$$

Of this 2,528 are required to heat the chimney gases, as has already been shown.

The generation of steam consumes as much as will evaporate 9.2 lbs. of water from 118° temperature of feed and at 115 lbs. pressure.

This requires $(1,219 - 118) = 1,101$ units per pound, or

$$1,101 \times 9.2 = 10,129 \text{ thermal units.}$$

Adding these we have 12,657 thermal units, and the difference, $14,340 - 12,657 = 1,683$ units, is lost by radiation from the exterior surfaces of the boiler and through the ash pit.

Summarizing we have:

Total heat corresponding to assumed composition of coal			15,038 thermal units.
Lost by incomplete combustion	657	heat units =	4.37 per cent.
" " evaporation of moisture	412	" " =	0.25 "
Absorbed by production of steam	10,129	" " =	67.36 "
Wasted by chimney gases	2,528	" " =	16.81 "
" " radiation, etc	1,002	" " =	10.21 "
	15,038		100.00

NOTES ON THEORETICAL CHIMNEY DRAUGHT OF "BERGEN."

The chimney has a cross-section of 14 square feet, and the cross-section of the tubes is practically the same. The average temperature of the gases in the chimney was 700° Fahr., and for the tubes we may assume a mean between 1,600, the probable temperature at entrance to the tube, and the 750° found in the uptake. This gives about 1,200° for the tubes. The coal burned per hour was 1,600 lbs., and the weight of gases 15.4 lbs. per lb. of coal.

Assuming 12.5 lbs. to the cubic foot, at 60° Fahr., and that the volumes are directly as absolute temperature as we have,

Velocity in chimney, 15.25 ft. per second.
" " tubes, 18.30 " " "

The resistance due to this velocity in the chimney, expressed in feet of the gas itself, is,

$$\frac{(15.25)^2}{2g} \times \frac{47 \times .012}{\text{Hydraulic mean depth.}} = \frac{(15.25)^2}{2g} \times 47 \times .012 = 4.1.$$

The hydraulic mean depth is unity, the chimney being oblong and 14 feet perimeter; 47 is the height of the chimney above the centre of uptake, and .012 a coefficient of friction. Similarly the head for the flues, which are $10\frac{1}{2}$ feet long, is found to be 2.5 feet. Reducing these numbers to their equivalent amounts for

500° * temperature, we have a total of 3.3 feet, which represents the length of chimney which would overcome the resistance of the flues and chimney.

To accelerate the gases there is required $\frac{(15.25)^2}{2g} = 3.7$ feet of gas at chimney temperature, and this reduced to 500° gives 3.1 feet of chimney.

For the combustion chamber enlargement and the right angle turn to the uptake, and obstruction due to damper, etc., we may, according to Péclet, allow about once the accelerating resistance.

For the resistance to passage through the grate a rule, in agreement with Rankine and Péclet, is that the feet of chimney equals the volume of gases, reckoned at 500° times the constant 0.0017.

This for the *Bergen* gives 49 feet.

Summarizing we, therefore, have:

Height of chimney to overcome acceleration	3.5 ft.
" " " " " friction of chimney	1.7 "
" " " " " " " tubes	1.6 "
" " " " " turns and damper	3.5 "
" " " " " resistance at grate	49.0 "
	59.3 ft.
Actual height	53.0 "

Considering the roughness of the constants, the agreement is apparently satisfactory, but when it is considered that the discrepancy is *a large fraction of all the resistances other than those due to the fire*, it must be concluded that the factor of greatest influence in the performance of a chimney, other than the damper, is the manipulation and condition of the fire.

FAILURE OF THE FURNACES OF THE "BERGEN."

The furnaces were cylinders, 36 inches diameter, $\frac{5}{8}''$ thick. To resist strains tending to collapse these cylinders, four heavy rings were riveted to each furnace, as shown in Fig. 87 and Fig. 86.

The rivets, near the grate level, which fastened a ring to the shell, gave evidence of their inability to avoid shearing off the rivet head lying within the furnace. The boat was, therefore,

* At this temperature the weight of gas is half that of atmospheric air at about 60°. So that the head of gas is then identical with height of chimney.

withdrawn from service on November 15, and corrugated or Fox furnaces are now replacing the original furnaces. It appears that the ring-braced furnace has given more or less trouble of the nature here described wherever it has been used, but that by using screw bolts instead of rivets, the rings have been kept in service. A probable hypothesis regarding the cause of the rupture of the rivets near a horizontal line is that the upper half of

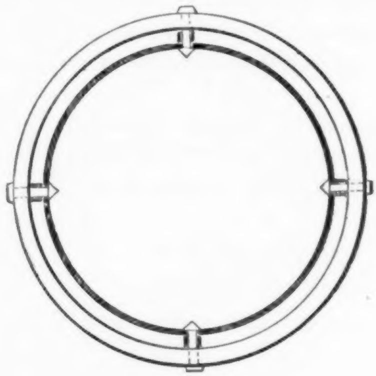


FIG. 87.

the circular furnace is at a sufficiently higher temperature than the lower half to cause it to expand, and tend to shear the horizontal rivets. In the case of the *Bergen*, 100° difference of temperature between the upper and lower halves of the furnace would cause sufficient expansion to depress the horizontal rivets $\frac{1}{32}$ of an inch on each side of the furnace. There is little doubt that such an amount of irritation might rupture rivets not having considerable surplus strength.

REMARKS REGARDING USE OF WATER-METER USED IN FEEDING BOILERS.

The use of a meter to record the water fed to boilers is the most convenient and best method of procedure if the instrument is reliable. A Worthington meter is practically a slide valve engine in its mechanism, and is as little liable to any sudden change in its condition as is a reliable engine. Without some such sudden change, a meter once carefully calibrated may be trusted to record feed water for a few days with more confidence than any human labor can be expected to operate in the weighing out and tallying of tanks.

Meters used for long periods finally become leaky, and thence arises the belief that they are unreliable.

If a meter is fitted in connection with a boiler so that it can be tested under the exact conditions of pressure and velocity used in feeding the boiler, the accuracy of the instrument can be definitely determined, and I have never known a Worthington meter to suffer any change of rate within a week of continuous use, even with water of 200° temperature.

In the cases of the *Bergen* and *Orange*, a special meter was constructed, through the kindness of Mr. C. C. Worthington, strong enough to stand 250 lbs. pressure. The meter was made of steel throughout, and had 3-inch inlet and outlet. It was used in the *Orange* on the feed-pipe of the pump, which was worked from the walking-beam. This pump was 7 inches diameter of plunger and 26 inches stroke. It was single acting, and ran at 25 revolutions per minute. The meter withstood several days of intermittent feeding with this pump, and suffered no injury. It was standardized by drawing off 25 cubic feet into a large tank resting on platform scales, and the rating obtained was 65 lbs. per cubic foot shown by the meter. The same meter was used on the *Bergen*, where it was supplied by a direct-acting steam pump. No difficulty was found in feeding against 160 lbs. boiler pressure. The calibration of the meter for the *Bergen* was made with 10 cubic feet portions, against the actual pressure and with the same rates of feeding as prevailed during the tests. The results varied from 63.8 to 64.4 lbs. per cubic foot. Less variation could be obtained by more elaborate trials, but no attempt was made to work closer than ½ lb. in 63 lbs., as other elements of discrepancy were of greater amount and uncontrollable—the variations of speed, steam pressure, and errors of indicators, for example.

TIGHTNESS OF VALVES AND PISTON OF "BERGEN."

No opportunity presented itself to test the degree of tightness of the valves and pistons. The high-pressure cylinder was slightly scratched. The other cylinders were in perfect condition so far as could be judged by the eye.

SPEED OF THE "BERGEN," USING DOUBLE SCREWS.

On September 18, 1889, the time of the *Bergen* from Seventy-second Street, New York city, to Newburgh, was 4 hrs., 54 min., 30 sec.

The distance travelled by shore was 55 statute miles. The speed along shore was, therefore, 11.22 statute miles per hour. The times of low water along the route and the tide-velocities were approximately as follows:

	Time of Low Water.	Time at which *Bergen* passed.	Tides. Statute miles per hour.
Seventy-second Street	10.15 A.M.	9.20 A.M.	− 1.5
Yonkers	11.00	10.30	− 1.5
Irvington	11.15	11.05	− 1.5
Haverstraw	11.30	11.40	− 0.5
West Point	12.15 P.M.	12.45 P.M.	− 0.5
Newburgh	12.45	2.14	+ 1.5

This would be equivalent to a tide of 0.7 miles against the boat for the whole distance, and would make the still-water speed about 11.92 miles. The speed by log was 12.88 miles, which, corrected, gives 11.88 as the still-water speed, but the correction for the log is not reliable within ½ mile per hour. The average revolutions of the engine were 142.4, and the slip was 16.1 and 16.6 per cent. respectively.

On the same trip the time between 10-mile posts at Riverdale and Irvington was 54 min. 21 sec., making a speed relative to the shore of 11.04 miles per hour. The probable tide against the boat was 1.5 miles, making the still-water speed 12.54 miles. The log gave 13.56 miles, which, corrected, makes 12.56 miles. The engine made 144 revolutions per minute, which makes the slip 12.8 per cent.

On the return trip from Newburgh the probable times of high water were as follows:

	Time of High Water.	Time at which *Bergen* passed.	Tides. Statute miles per hour.
Governor's Island	3.00 P.M.		
Seventy-second Street	3.30	7.10	+ 2.5
Riverdale	4.30	6.11	+ 1.0
Yonkers	4.45	5.55	0.
Irvington	5.00	5.22	− 1.5
Haverstraw	5.15	4.45	− 2.5
West Point	6.00	3.45	− 2.0
Newburgh	6.30	2.15	− 1.5

This is equivalent to an average tide of 0.8 mile against

the boat. The whole distance was 49* miles in 3 hours, 58 minutes.

This gives a speed with reference to the shore of 12.32 statute miles, or 13.14 miles through the water. The engine made 145 revolutions, whence the slip was 9.4 per cent. The time between 10-mile posts on the return trip at 5.22 P.M. was 49 minutes, making a speed of 12.25 statute miles with reference to the shore. The tide averaged 0.36 mile against the boat, making the still-water speed 12.61 statute mile per hour. The log showed 12.72 miles. The engine ran at 145 revolutions, making the slip 12.8 and 11.5 per cent. respectively.

Summarizing, we have :

Description of Course.	Speed Statute Miles per hour.	Slip per cent.	Revolutions per minute.	Boiler Pressure, lbs. per sq. inch. Gauge.	Condition of Throttle.	H.P.
55-mile run to Newburgh....	11.92	16.2	142	114	Wide open.	662
	11.88	16.6	142	114		662
10 " " via "	12.54	12.9	144	114		700
	12.56	12.8	144	114		700
49 " " from "	13.14	9.5	145	120	Partly closed.	667
10 " " via " ...	12.61	12.8	145	120		700
	12.72	11.5	145	120		700
Average..................	12.49	13.2	144			

The 49-mile run may be rejected on the ground of too much irregularity in the conditions.

During the return trip from Newburgh, Sept. 18, short runs were made at 71 and 114 revolutions, respectively, and the speed determined by the log only to be 6.4 and 10.5 miles per hour, respectively, the corresponding horse-powers being 97 and 304, and the slip about 10 per cent. These figures must be regarded as very rough approximations. Very light winds and smooth water prevailed during the experiments.

On March 30, 1889, a run was made over a one-mile course at Spuyten Duyvil, the engines developing 1,007 H. P. by allow-

* The engines averaged 145 revolutions until 6.30 P.M. They were then run 25 minutes at 71.3 turns per minute, and then 10 minutes at 113 revolutions; during these periods the distance run, as determined by the corrected log readings and the tide table, was 6 miles, which is subtracted from the distance to Seventy-second Street, making 49 miles.

ing the boiler pressure to accumulate just before making the run. The data were as follows:

Time to run one mile against tide....................	5 mins. 18 secs.
Revolutions per minute..........................	156.6
Time to run one mile with tide..........................	3 mins. 21 secs.
Revolutions per minute............................	168.6

Placing this data in the following equation:

$$(1 - \text{slip}) \times \begin{pmatrix}\text{pitch of}\\ \text{screw}\end{pmatrix} \times \begin{pmatrix}\text{revs. per}\\ \text{minute}\end{pmatrix} \times \begin{pmatrix}\text{time of}\\ \text{run}\end{pmatrix} + \begin{pmatrix}\text{velocity}\\ \text{of tide}\end{pmatrix} \times \begin{pmatrix}\text{time of}\\ \text{run}\end{pmatrix} = 1 \text{ mile},$$

the *plus* sign being used for the run with the current, and the *minus* sign for the opposite run; we have two equations, from which the slip and tide is determined to have been 11 per cent. and $2\frac{3}{4}$ statute miles, respectively. The time of the test was during the period of high ebb velocity. We therefore have the still-water speed 14.07 against the tide, and 17.9 with the tide, or a mean speed of 14.6 miles for 163 revolutions per minute, and 1,007 H.P.

SPEED OF "BERGEN," USING ONE SCREW.

On September 28, 1889, trials were made over a two-mile course at Spuyten Duyvil, with one screw removed, and the remaining screw worked first at the stern and then at the bow of the boat. The results are shown in Table X.

The tides prevailing during these experiments are shown in Fig. 80. The observations were made from two boats anchored at the north and south extremities of the course, respectively, by allowing a submerged float to drift astern by means of a thread unwound from a reel located in the boat. It will be noticed that in the two north runs made with the screw astern the slip is but about two-thirds of that for the run south. It is thought that this discrepancy is due to the fact that the tide differed about a mile an hour between the two ends of the course, being greatest at the north end. Hence, in running north, the boat was moving through water which may be conceived as having an increasing velocity, averaging half a mile per hour, so that the screw would tend to have its slip diminished by the effect of this velocity. These two cases are, therefore, discarded for purposes of comparison of still-water speeds. The

TABLE X.

SPEED TRIALS OF "BERGEN," WITH ONE SCREW IN USE, SEPTEMBER 28, 1889.

Time.	Direction of Boat.	Interval by Stop Watch. Minutes.	Speeds. Statute miles per hour. By Log.	Relative to shore.	True.	Tide. Average by No. and So. Boats. Miles per hr.	Excess of Log Reading.	Boiler Pressure. Gauge. Lbs. per square inch.	Revolutions per minute.	Mean Effective Pressures. Lbs. per square inch. High cylinder.	Intermediate cyl'der.	Low cylinder.	Horse Powers. High cylinder.	Intermediate cylinder.	Low cylinder.	Total.	Slip of screw per cent.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
								SCREW ASTERN.									
A. M.																	
11.25	1 South	11.583	13.41	10.36	12.10	1.75	1.31	77	148								18.3
11.55	1 North	8.49	13.21	14.13	12.49	1.64	0.72	65	142								12.1
										27.8	15.5	6.12	124.7	150.5	145.0	420.2	
P. M.																	
12.19	2 South	11.3	13.36	10.62	11.98	1.36	1.38	66	145	29.25	17.5	6.12	134.0	173.6	148.2	455.8	17.4
12.53	2 North	9.58	12.60	12.5[illegible]	12.08	0.45	0.52	59	138	26.25	15.75	6.0	113.6	147.6	143.2	404.4	12.6
1.15	3 South	10.17	12.72	11.8[illegible]	11.8	0.	0.92	67	146	27.5	17.5	6.5	126.9	174.8	158.4	461.0	19.0
								SCREW AHEAD.									
1.46	1 North	10.24	12.13	11.72	11.70	0. *	0.43	70	148	31.75	19.5	6.75	147.5	196.0	165.5	509.0	21.2
2.19	1 South	10.15	11.90	11.82	10.8	1. *	1.10	68	144	27.5	17.5	6.62	125.0	172.4	159.2	456.6	25.0
2.39	2 North	11.63	12.10	10.32	11.57	1.25*	0.53	70	145	28.25	20.0	6.5	132.6	198.4	157.3	488.3	20.2
3.06	2 South	9.44	12.43	12.71	10.96	1.75*	1.47	70	146	27.9	17.9	6.45	128.7	158.7	155.9	443.3	24.8
3.25	3 North	12.82	12.11	9.36	11.36	2.00*	0.75	67	142½	27.0	17.5	6.0	121.6	170.6	143.0	435.2	19.7
3.44	3 South	9.033	12.18	13.29	11.19	2.10*	0.99	70	146	28.5	18.0	6.25					24.4
4 09	4 North	13.55	12.09	8.86	11.11	2.25	0 98	70	144	28.75	16.0	6.12					22.7
4.27	4 South	8.667	12.10	13.85	11.55	2.3	0.55	70	146	28.25	17.5	6.0	130.3	174.8	146.2	451.3	20.2
								SCREW ASTERN.									
4.56	1 North	11.067	14.84	10.84	13.40	2.56	1 44	94	164	39.5	21.75	8.56	204.7	244.0	234.3	683.0	18.1
5.10	1 South	7.55	14.08	15.90	13.40	2.50	0.68	90	162	40.25	22.0	8.75	206.0	243.8	236.6	686.4	17.3
5.46	2 North	26.8	7.91	4.48	6.98	2.50	0.93	110	83	11.75	6.5	1.69	30.8	36.7	25.1	92.6	16.0

* Tides, deduced from "mean tide curves," Fig. 80.

Except during the last experiment, the throttle was wide open

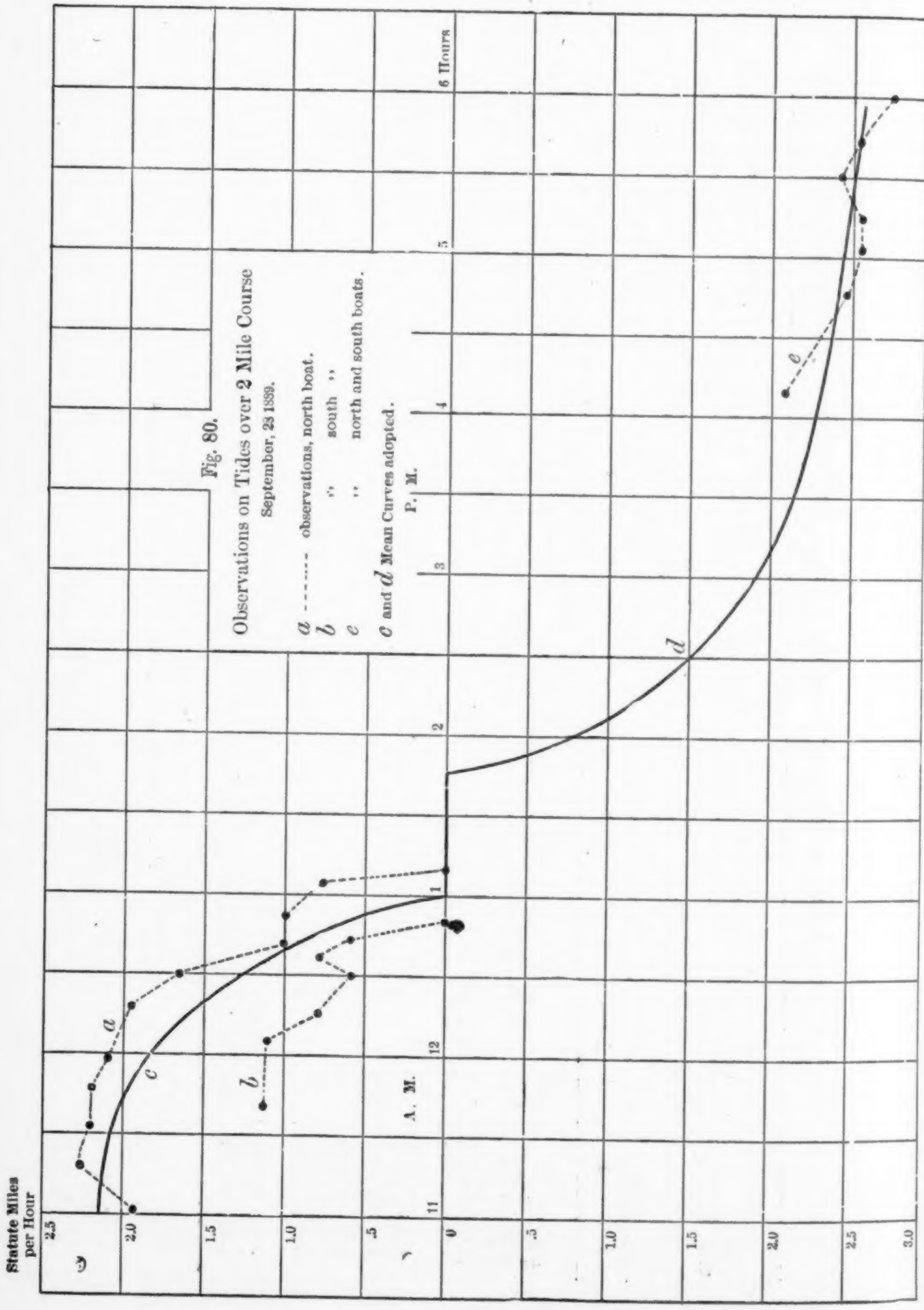


Fig. 80.
Observations on Tides over 2 Mile Course
September, 23 1889.
a ------- observations, north boat.
b " " south "
e " " north and south boats.
c and d Mean Curves adopted.
Statute Miles per Hour
A. M.
P. M.
6 Hours

Tide Curves.

TABLE XI.

TRIAL OF "BERGEN" IN FERRY SERVICE ON HALF-HOUR TIME BETWEEN HOBOKEN AND BARCLAY STREET, NEW YORK CITY, OCTOBER 10 AND 11, 1889.

Time.					Revolutions per Minute.		Steam Pressure. Pounds per Square Inch—Gauge.				Vac. in Inches.	Delays.
Leave.		Total Time at Full Speed.	Total Time to Arrive at Slip.	Interval of Safety Valve Blowing.	At Full Speed.	While Working into Slip.	Boiler.		Receivers.			
							Leave.	Arrive.	First.	Second.		
1	2	3	4	5	6	7	8	9	10	11	12	13
	P. M.	M. S.	M. S.	Secs.								
New York...	7.20.30	7.30	9.30	60	141	52	146	141	35	2.0	27	
Hoboken....	7.35	8.30	11.00	70	136.4	39	145	142.5	36	2.25	28.25	
New York...	7.50	7.00	10.00	145	144	38	144	141	34	2.0	25.25	
Hoboken....	8.05	8.30	11.00	200	163	40	146	137	35	2.5	27	
New York...	8.19	7.00	9.00	280	142	36.5	140	136	34	2.25	26.75	10 sec.
Hoboken....	8.35	9.00	11.30	....	139.4	39	141	132.5	35	3.0	27.75	
New York...	8.50	6.30	9.00	....	148	38.4	135	133	33	2.0	27.0	
Hoboken....	9.17	9.00	12.00	....	137	43.3	140	136	34	1.0	29.5	
New York...	9.33	6.00	8.00	....	150	92.0	142.5	134	32.5	1.5	26.5	
Hoboken....	9.48	8.30	11.30	....	144.4	42	145	133	34	2.25	27.0	
New York...	10.03	7.00	9.00	....	140	56.5	144	145	34	1.75	27.0	
Hoboken....	10.18	9.00	12.00	....	136.3	33.3	145	146	38	3.00	27.0	
New York...	10.32	7.00	10.00	75	145	34	145	145	37	2.0	27.0	
Hoboken....	10.46.30	9.30	12.30	....	127.3	42	145	139	38	3.0	27.25	
New York...	11.02.0	7.00	9.00	....	146	55.5	145	145	38	2.25	27.25	
Hoboken....	11.17	10.00	12.00	....	117.5	35	145	140	37	2.75	27.0	
New York...	11.32	8.00	10.00	....	129.1	68	141	135	34	1.5	27.75	1 min.
Hoboken....	11.48	8.00	11.00	....	139	36.3	141	137	34	2.25	27.5	
New York...	12.02.30	7.30	10.00	....	151	53	136	127	35	2.0	27.25	
Hoboken....	12.18	7.30	10.00	....	140	52	139	135	37	2.75	28.25	
New York...	12.32	7.30	10.30	....	150	41	138	134	35	2.00	28.25	
Hoboken....	12.47.30	8.00	10.30	....	130.2	43	141	140	35	2.75	27.5	
New York...	1.03 a.m.	7.00	9.30	....	166	44.4	140	131	35	2.0	28.0	
Hoboken....	1.17.0	7.00	9.45	....	132.1	46.5	140	131	33	2.0	29.0	
New York...	1.34.25	8.10	11.00	....	159.2	34.5	138	140	40	4.0	28.0	
Hoboken....	1.49.0	5.24	8.45	....	154.6	26.5	145	139	30	3.5		
New York...	2.01.50	8.25	10.40	75	152.4	46	144	120	35	2.5	28.0	
Hoboken....	2.33.10	7.35	10.35	....	127.5	46.3	136	130	32	3.0	28.5	
New York...	2.47.0	6.35	11.50	....	165	23	136	125	30	1.5	29.25	
Hoboken....	3.03.35	7.55	10.35	....	140	48	140	121	81	3.0	29	30 sec.

TABLE XI.—*Continued.*

Time.					Revolutions per Minute.		Steam Pressure. Pounds per Square Inch—Gauge.					
Leave.		Total Time at Full Speed.	Total Time to Arrive at Slip.	Interval of Safety Valve Blowing.	At Full Speed.	While Working into Slip.	Boiler.		Receivers.		Vac. in Inches.	Delays.
							Leave.	Arrive.	First.	Second.		
1	2	3	4	5	6	7	8	9	10	11	12	13
	P.M.	M. S.	M. S.	Secs.								
New York ..	3.17.15	9.30	12.15	....	145.4	49.4	130	120	30	2.0	28	
Hoboken....	3.32.25	7.00	10.00	....	141	33.3	133	133	33	1.75	29.25	
New York...	3.47.15	9.45	12.10	....	142.2	62	145	127	36	3.0	28.5	
Hoboken....	4.02.35	7.05	9.13	....	133	89	145	142	35	2.0	28.75	
New York...	4.16.00	10.00	11.50	....	141.3	57	145	135	36	2.0	28.0	
Hoboken....	4.31.20	10.12	12.55	....	80.1	42.3	146	148	25	2.5	28.75	3 min. 40 sec.
New York...	4.46.45	9.45	11.45	120	142	42.5	147	135	32	1.0	28.0	
Hoboken....	5.02.45	8.40	10.00	....	115	99	141	130	35	2.25	28.5	
New York...	5.15.50	9.25	11.25	....	140	54.5	130	111	35	1.5	28.5	
Hoboken....	5.32.10	8.10	10.35	....	129	48	121	120	30	1.0	28.75	
New York...	5.46.20	8.30	10.50	....	141.1	48	131	124	32	1.0	28.5	
Hoboken....	6.01.40	7.50	10.05	....	133	42	131	119	32	1.5	29.0	
New York...	6.20.40	7.45	10.55	....	133	67	121	110	27	−2.0	29.5	
Hoboken....	6.41.35	8.25	10.39	....	134	49	141	130	32	1.5	29.5	
New York...	7.00.42	7.18	9.53	....	141	40	141	123	32	1.0	27.75	
Hoboken....	7.23.05	9.19	11.25	....	128.5	46	140	126	31	1.0	29.0	
New York...	7.42.00	6.35	9.15	....	143	45.4	138	122	32	1.0	27.0	
Hoboken....	8.01.45	9.15	11.25	....	133.1	42	136	115	32	1.5	28.0	
New York...	8.21.11	7.09	9.16	....	136	47.2	130	114	28	0.25	27.5	
Hoboken....	8.43.00	9.30	11.30	....	137.2	45.5	135	122.5	20	0.25	29.0	
New York..	9.08.00	6.00	8.30	....	158.0	61.0	140	132.5	35	2.25	24.5	
Hoboken ..	9.22.00	9.30	14.00	....	118.3	51.0	140	135	30	1.25	29.0	3 min.
New York...	9.41.00	9.00	11.30	....	112	42.4	142	125	34	2.00	27.5	
Averages...............		8.01	10.35		139.4	44.7	139.5	129.6	33.6			

Engine was at rest while boat was in slips.

conclusions to be drawn from the experiments are shown in the following table:

TABLE X. (*a*).

	Revolutions per minute.	H.P.	Still-water Speed. Statute miles per hour.	Slip, Per cent.	Winds.
Screw at Stern . . .	145	458	11.96	18.2	Light Westerly.
" at Stern......	163	684	13.42	17.7	
" at Stern......	83	93	6.98	16.0	
" at Bow.......	145	451	11.28	22.3	

PERFORMANCE OF "BERGEN" IN FERRY SERVICE.

On October 10, 1889, observations were made on the running time during 14.3 hours, commencing at 7.20 P.M., the boat making half-hour time between Hoboken and Barclay Street, New York City; these are shown in Table XI.

The average interval during which the engines were at full speed was 8 minutes; allowing 20 seconds for the boat to acquire full speed after the engines were at full speed, and 300 feet for distance from slip at slow-down bell, it follows that the boat ran 1.65 statute miles in $7\frac{2}{3}$ minutes, or at the rate of 12.9 miles per hour.

The steam pressure averaged 140 lbs., but the engines were throttled to 100 lbs. admission pressure, and the link used to shorten the cut-off to a slight extent. The boilers were run with the damper considerably closed, to prevent blowing at safety valve while in the slip. When at full speed the revolutions averaged 140 per minute, and the indicator cards were as per Fig. 76. While working into the slip the revolutions averaged 45 per minute, and the indicator cards were as per Fig. 77. The horse-power at full speed was 650; while working into the slip, 45, and the revolutions 45 per minute.

The total time spent at full speed was 7.3 hours. The total time working into slip was 2.08 hours. The latter is equivalent to $\frac{45}{140} \times 2.08$ hours at full speed, but the consumption per stroke at slow speed is only about $\frac{40}{120}$ of that for full speed,—taking the consumption proportional to the absolute pressures of admission,—so that the slow speed consumption is on the whole equivalent to $\frac{45}{140} + \frac{40}{120} \times 2.08 = 0.22$ hours at full speed.

Hence the steam consumed in the whole interval was devoted to

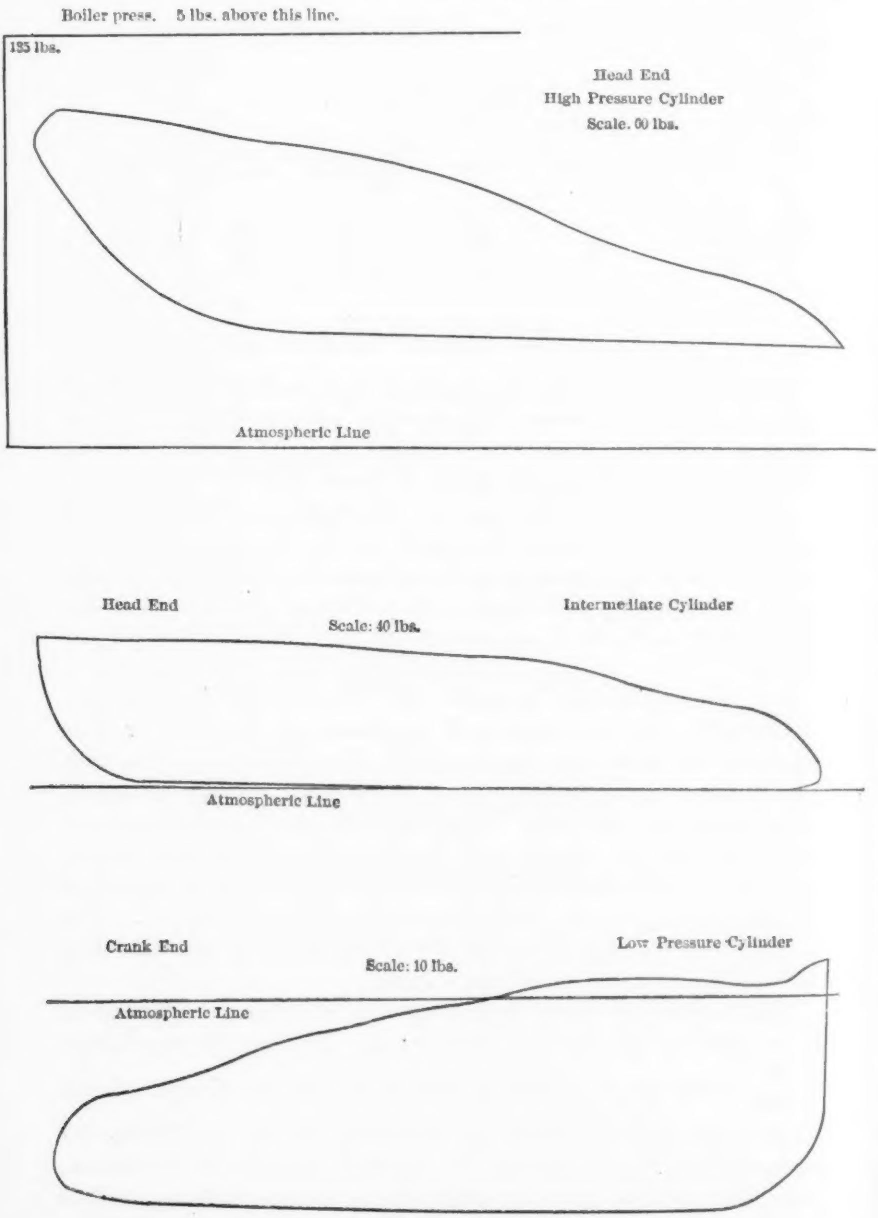


FIG. 76.

Average Card of the *Bergen* in Regular Ferry Service. Full Speed, Steam Throttled 35 H.P., 650.

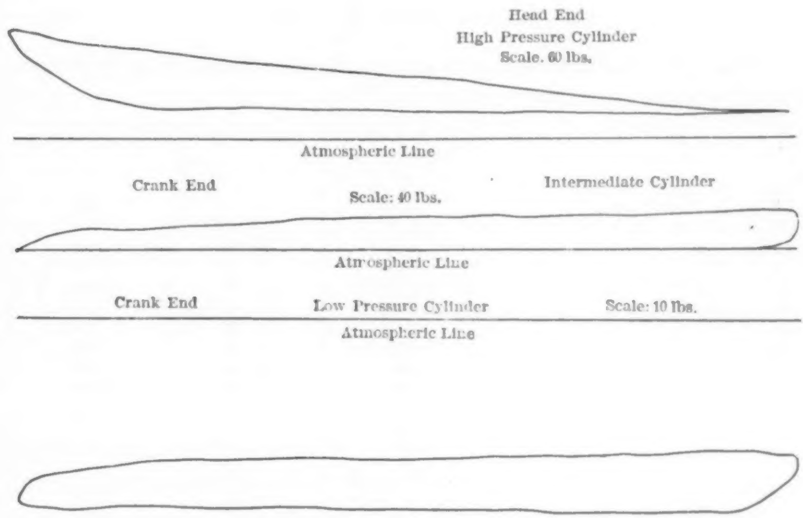


FIG. 77.
Average Cards of the *Bergen* in Regular Ferry Service while Working into the Slip. Steam Throttled, 100 Lbs.

the exertion of 650 H.P. for $7.3 + 0.22 = 7.55$ hours. The steam consumption for the whole interval of 14.3 hours was 7,892 lbs. per hour. The engine was running at full power $\frac{7.55}{14.3} = 0.528$ of the whole time. Hence the 650 H.P. during 7.55 hours must be conceived as equivalent to $650 \times 0.528 = 344$ H.P. for the whole time. The consumption per hour per H.P. is, therefore,

$$7{,}892 \div 343 = 22.9 \text{ lbs.}$$

This exceeds the result in steady service by 1 lb., or about $4\frac{1}{2}$ per cent.

The evaporation of the boilers was determined over an interval of 26 hours of continuous service, and found to be 8.4 lbs. of steam per lb. of coal.

The rate of combustion was 10.5 lbs. per hour per sq. ft. of grate surface.

THEORETICAL SPEED OF "BERGEN."

The immersed surface and angles of water lines are as follows :

	Sq. ft. Surface.	Maximum Angle.	Co-ef. of Aug. $1 + 4 \sin^2 + \sin^4$.
Keel and Rudders.............	400	0°	1.000
Keel to 1 ft. line...	248	2° 40′	1.010
1 ft. " 2 " "	620	5° 50′	1.040
2 " " 3 " "	620	8° 50′	1.094
3 " " 4 " "	620	10° 50′	1.143
4 " " 5 " "	620	13° 10′	1.210
5 " " 6 " "	620	16° 30′	1.326
6 " " 7 " "	620	20° 30	1.505
7 " " 8 " "	682	26° 50	1.848
8 " " 9 " "	732	28° 20	1.940
Total....................	5,788	Av. = 13°.13′	Av. = 1.299

Multiplying the average of each pair of co-efficients by the corresponding surface, and dividing the sum of the products by the total surface, we find the mean co-efficient of augmentation to be 1.299. The augmented surface is, therefore, 7,524 square feet.

Column 5 of Table III. shows that the speeds are practically proportional to the cube roots of the horse-powers over a large range * of power and engine speed. Column 6 is computed by the formula :

$$\text{Speed statute miles per hour} = 1.15 \sqrt{\frac{21,200 \text{ H. P.}}{\text{Aug. surf.}}}, \qquad (1)$$

* Inasmuch as both the speed and mean effective pressures are simultaneously reduced as the indicated horse-power decreases, it is probable that the friction of the machinery is nearly proportional to the indicated horse-power throughout the range covered by the table. In this respect marine engines differ from the case of stationary engines, the friction of which has been shown to be constant over a large variation of indicated horse-power. This is explainable on the following grounds : In the case of the stationary engines, the variation of horse-power is caused by variation of mean effective pressure only, and the variations of power as measured by a prony break do not largely affect the pressure of the main shaft upon its bearings. The friction of the latter, which is the largest item in the total friction of the engine, is not therefore much affected by variations of power (see discussion paper 316, Vol. X. Trans. A. S. M. E.) ; but in the case of the marine engine the velocity of the point of effort being a less multiple of the piston speed, the variations of pressure upon the bearings of the main shaft are a larger fraction of latter's dead weight ; and furthermore, by the variation of speed accompanying variations of power, the entire dead weight friction varies rapidly with variations of power.

in which the constant 21,200 represents an allowance of 20 per cent. of the indicated power for losses by friction in mechanism and at the screw, and 16 per cent. of the net power in slip. The allowance of 20 per cent. for friction is satisfactorily checked by computing the friction of the various parts of the engine, allowing a co-efficient of friction of 0.10. The allowance of 16 per cent. for slip makes the latter correspond to the average slip for one and two screws respectively. The most probable slip for two screws is taken at 12.6 per cent., and that for one screw at 18 per cent. The extreme variation of the constant, 21,200, between these limits affects the speed by only about $1\frac{1}{2}$ per cent., which is within the limit of other causes of discrepancy. The net power devoted to the propulsion of the hull only is therefore :

$$(1 - .20)\,(1 - .16) \times \text{H.P.} = .67\ \text{H.P.}$$

Column 7 is estimated from the formula :

$$\text{Area screw disk} \times \frac{63}{32} \times \frac{(\text{speed of vessel})^3}{(1 - \text{slip})^2} \times \frac{\text{slip}}{550} = .67\ \text{H.P.}, \quad (2)$$

in which the speed of vessel is in feet per second. This formula is that given by Rankine when the difference between real and apparent slip is neglected. The lines of the *Bergen* would make such difference less than 2 per cent., if anything, but the results in Column 7 are already too large, and would be made larger by assuming any slip greater than the apparent slip. The excess of these results over the true speed may be due to too small a value of apparent slip, or to the fact that the effective area of the steam acted upon should be taken greater than the disk of the screw itself. Rankine indicates that such is the case, and the increase of area necessary to bring the figures in Column 7 into agreement with those in Column 3 agrees with that given in his "Ship Building," namely, about $\frac{1}{4}$ of the disk area.

LOSS OF EFFECT OF BOW SCREW.

Column 4, Table X. (*a*) shows that the screw at bow propelled the boat about $\frac{5}{8}$ of a mile slower at 145 revolutions than did the same screw acting at the stern. As the horse-power expended was the same, it is probable that the less speed in the former case is due to the recoil of the water from the bow screw against the vessel's hull. The maximum effect of such recoil may be conceived as causing an increased resistance equal to that re-

quired to propel a portion of the boat's immersed surface with an increased velocity equal to the slip.

If the portion of the boat's surface taken be that included in a draught equal to the diameter of the propeller, this surface will be about 4,374 sq. ft., with an average co-efficient of augmentation equal to 1.166, making an augmented surface equal to 5,100 sq. ft., which will move, with reference to the water, with a velocity equal to

$$\text{speed of boat}\left(1+\frac{0.18}{1-0.22}\right)$$

0.22 being the apparent slip of screw at bow, and 0.18 its probable real slip, taken equal to the slip for screw at stern at equal revolutions per minute. The resistance of this portion of the boat will, therefore, be proportional to $v_1^2\left(1+\frac{0.10}{1-0.22}\right)^2 \times 5{,}100$, v_1 being the actual speed of boat with screw at bow.

The remaining portion of the boat's immersed surface will be 1,414 sq. feet, with 1.394 for the co-efficient of augmentation. The resistance of this portion will be proportioned to

$$v_1^2 \times 1{,}414 \times 1.394 = v_1^2 \times 1{,}973.$$

The power expended will be proportional to the product of the sum of these two resistances by the speed of boat, or to

$$v_1^3\left[\left(1 \times \frac{0.18}{1-0.22}\right)^2 \times 5{,}100 + 1{,}973\right]$$

With the screw at the stern the power is proportional to

$$v^3 \times 7{,}500,$$

in which v is the still-water speed, and 7,500 the total augmented surface.

Since the power expended for screw at either bow or stern is by Table X. (a), practically the same, we have, by equating the above expressions,

$$v_1 = \sqrt[3]{\frac{7{,}500}{1{,}973 + 5{,}100\left(1 \times \frac{0.18}{1-0.22}\right)^2}} = 0.95\ v.$$

Taking v at 11.96 statute miles, as per Table X. (a), we have $v_1 = 11.3$ statute miles. The actual speed was 11.28, or practically the same.

It is, therefore, concluded that the bow screw is less effective than the one at the stern by the extent to which the water

thrown astern by its slip increases the velocity of rubbing between the boat and the water.

EXTRA HORSE-POWER REQUIRED BY TWO SCREWS.

It will be noticed that by Tables X. (a) and III. one screw is apparently able to propel the boat with less horse-power at a given speed, than are the double screws. Thus, one screw at stern, making 163 revolutions per minute, drove the boat 13.42 miles with 684 H.P., whereas two screws at 145 revolutions required 700 H.P. for a speed of 12.62 miles. Applying the law of cubes, and allowing an error of 3 per cent. in the horse-powers in opposite directions, we find that the probable facts might be as follows:

	Revs.	H. P.	Speed.	Slip.
Two screws.................	145	679	12.62 miles	12.6%
One screw at stern	152	584	12.62 "	18.0%

If now we write the following two equations on the basis of formulæ (1) and (2):

H.P., two screws, including power lost by slip,

$$= \frac{\overline{1.68}^2 \times 550}{0.67} \times \frac{v^3 \times s \times 4 \times \text{disk of screw}}{(1-s)^3} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$\text{H.P., one screw} = \frac{\overline{1.68}^2 \times 550}{0.67} \times \frac{v^3 \times s' \times 2 \text{ disk of screw}}{(1-s)^3}. \quad (5)$$

Then, by equating the right-hand members, and assuming v equal in each, if $s' = 18$ per cent., s should equal $11\frac{1}{2}$ per cent. In other words, unless the slips of the screws are related in the ratios of 18 to $11\frac{1}{2}$, the horse-power for equal speed will not be the same. By the above data the slip is in the first instance 12.6 per cent. for the two screws, and substituting this in formula (4), it results that the horse-power should be 15 per cent. more than that for one screw, as given by formula (5) for 18 per cent. slip. This would make the horse-power 689 for the double screws, against 679 actual. It appears, therefore, that even neglecting the friction of the screw-blades in water, the two screws should require more power than a single screw having the same diameter as each of the double screws, a result due to the recoil of the slip water, as explained above. So far as we now know, the loss by friction between the screws and the water is only about 2 per cent. of the power exerted by the screw.

CONDITIONS OF WETTED SURFACES.

Both the surfaces of the screws and of the boat were cleaned three days before the trials of the single screw. These surfaces were also but slightly foul during the experiments with two screws.

LONG SERVICE TRIAL OF "BERGEN" VERSUS "ORANGE."

During October, 1888, the *Orange* was operated steadily, on the Barclay Street route, 339 hours, averaging about 15 hours daily. Her coal consumption, including fuel for banking and starting fires, was 1,027 lbs. per hour.

During June, 1889, the *Bergen* was operated the same number of hours, and upon the same time-table as the *Orange*. Her coal consumption was 936 lbs. per hour. The coal used by the *Bergen* was, therefore, about 9 per cent. less than that used by the *Orange*.

By the results given in Tables II., VII. and XI. for the 14-hour ferry service test, it appears that, supposing the boilers of both boats equal in economy, the *Orange* should consume 26 lbs. of steam for each H.P. and develop 810 H.P. in making a certain distance in seven minutes, while the *Bergen* should consume 23 lbs. of steam and make the same distance with 650 H.P. in $7\frac{2}{3}$ minutes. The consumption of the *Bergen* was, therefore, $\frac{23 \times 650}{26 \times 810} = 0.78$ of that of the *Orange*. In other words, the *Bergen* used 22 per cent. less coal than the *Orange*. The less saving during the 339 hours' trial is easily explainable on the following grounds:

1st. During 339 hours' run, the *Bergen's* engine was run nearly the whole time the boat was in the slip; while during the 14 hours' test the engine was, by special instructions, not run while in the slip. The pumps of the *Bergen* exhausted overboard during both tests, and were run in the slip. 2d. The *Orange* is not required to develop 810 H.P. during her operation on the day service which consumed the 339 hours' test, whereas the *Bergen* was being run to the full limit of speed allowed under the engineer's instructions. This speed, which was 145 revolutions, would develop 665 H.P. Hence, the coal consumption should have been nearer equal to that of the *Orange* during the 339 hours' test.

RELATIVE ECONOMY OF "BERGEN" AND "ORANGE" AT 14 MILES PER HOUR IN FERRY SERVICE.

When the *Orange* exerts 810 H.P. and makes the full speed distance between Hoboken and Barclay Street in 7 minutes, she travels 14 miles per hour through the water and uses the full capacity of her boilers, which then generate 500 H.P. during the whole time available—including both the running time and time in slip.

To make the same speed, the *Bergen* must reduce her time of $7\frac{2}{3}$ minutes, requiring 650 H.P., to 7 minutes, which requires her engines to exert

$$\left(\frac{7\frac{2}{3}}{7}\right)^3 \times 650 = 1.4 \times 650 = 910 \text{ H.P.}$$

As the time this H.P. must be exerted is but about half of the time which is available to the boiler for generating the steam, the capacity of the boilers need be but 455 H.P., whereas the run to Newburgh showed them capable of maintaining 665 H.P. Hence the *Bergen* has ample boiler capacity to maintain the required 910 H.P. Her coal consumption, assuming the pumps etc., to exhaust into the main condenser, will be proportional to $910 \times 22 = 20{,}020$, and that of the *Orange* to $810 \times 26 = 21{,}060$. This leaves a margin of 5 per cent. in favor of the *Bergen*.

POSSIBLE MAXIMUM ECONOMY OF A SCREW-BOAT VERSUS PADDLE-BOAT.

Assuming that three-fourths of the pump's, etc., consumption can be saved, and that the full advantages of the triple expansive system can be made available by maintaining 150 lbs. pressure, the steam consumption per hour per H.P. of all the machinery may be safely estimated at 18 lbs. This would make the consumption relative to the *Orange* for 14-mile speeds, as $910 \times 18 = 16{,}380$ to 810×26, or as 78 to 100, making a margin of 22 per cent. in fuel. The same size of engine as in the *Bergen* could command 1,000 H.P. with about $\frac{2}{3}$ of the present boiler capacity. This will make the boilers of a screw-boat weigh about the same as those of the *Orange*.

The total weight of the screw-boat will be about 220,000 lbs. lighter, 80,000 lbs. of which is due to absence of paddle wheels, and 140,000 lbs. to difference of weight of hull. The difference of first cost due this difference of weight plus the above saving of coal, must be put against the extra repairs of the machinery and extra attendance of the screw-boat. Prolonged experience with

the new type of boat can alone settle exactly the balance in its favor, but that there will be a considerable balance financially there need be little doubt, while the advantages of better accommodation, greater attractiveness, and greater control in service are incontestable.

DISCUSSION.

Mr. C. E. Emery.—I desire to express my appreciation of the thorough manner in which the experiments referred to in this paper have been made, and also wish to compliment the public spirit of the Ferry Company and of Col. Stevens, representing it, for putting at the disposal of gentlemen competent to make a series of experiments of this kind, this novel vessel for experimental purposes.

In examining the results of the experiments, I am surprised at the very low economy shown by the triple compound engine. I would ask what was the mean pressure referred to the large cylinder?

Prof. Denton.—Twenty seven pounds.

Mr. Emery.—That is ample, so that the low economy is difficult to account for. There are plenty of experiments showing that a horse-power can be obtained in a compound engine of good construction with 18½ or, at the most, 20 pounds of feed water per horse-power per hour. In a recent paper on the "Development of the Compound Engine," I predicted from the results of experiments with the revenue steamer *Rush* that, by doubling the pressure, a horse-power could be produced in a triple compound engine for 15.15 pounds of feed water per horse-power per hour, and the accuracy of this prediction has been shown quite recently. It is a question, therefore, why the results with the triple compound engine referred to in the experiments now presented were so very inferior.

Prof. Denton.—It is possible there was some leakage of the valve of the *Bergen*. We did not have a chance to try the leakage of the valve, but the economy is proportional to that of the *Meteor*, taking differences of expansion and cushion into account. I want to say, regarding the *Orange*, inasmuch as there is a great deal of assertion about the possible leakage of poppet valves, and as is well known the leakage varies with the tendency of the steam chest to expand out of shape, we went to the trouble of testing very thoroughly the leakages of the valves and pistons.

The greatest leakage was one-fourth of one per cent. The piston had no leakage at all, and yet it is packed with a ring of its own elasticity, having about 5 pounds pressure to the square inch.

Mr. Emery.—Engines running with 70 pounds—compound engines—have given 13 pounds of water per indicated horse-power per hour; and so, if the triple compound is not going to do better than that, there is no necessity of having any extra complication.

Mr. Wm. Kent.—I would ask if the very large clearance is not largely responsible for the lack of economy; also the great drop between the boiler pressure and the steam-chest pressure.

Mr. C. A. Hague.—It would seem as though the large clearance would affect the economy of the engine somewhat, and I would like to ask, why, in the triple expansion engine slide valves are not used?

Col. Stevens.—Although I am not personally an engineer, I can tell you, perhaps, that Mr. Wilson's idea in designing the engine was that the piston valve would give, on the whole, the best results; that with excessive pressures, which might be brought on the back of the slide valve, he would prefer the larger clearance necessary for the piston valve. Also, to obviate any question of leak in the piston valve, he placed the piston valve between the high pressure and intermediate cylinders.

A Member.—If the gentlemen have got through with the question of engine, I would like to know very much what is the matter with the boiler. The professor explained it, that those rivet heads come off from the downward movement of the ring. I would like to ask how that is likely to take that head off the rivet. If the boiler is under high pressures, and circulation is good, why should not that boiler hold? Is there any reason why that ring there should collapse? I would like to have somebody's opinion on that.

A Member.—Do those occur only in the hottest parts of the furnace, or right along?

Col. Stevens.—I think probably it would be on the hottest parts of the furnace only. I think it is principally on the hottest parts of the furnace. The trouble is principally in one furnace. It occurred in both furnaces and one boiler.

Prof. Denton.—I inquired of the master mechanic this morning, and he said that several rings along the boiler had shown

this same weakness, and I have understood that the rivets were essentially horizontal rivets, or near the horizontal line. Of course, if that is not so, why, any hypothesis such as mine falls to the ground.

Mr. Emery.—What was the thickness of the shell?

A Member.—I think nine-sixteenths.

Mr. Hague.—Is there any evidence of the rivets burning off? Is that a water space in between the two rings?

Prof. Denton.—The ring is right in the water; the rings do not enclose a place. They are four feet apart, just like the hoops on a barrel.

Mr. Hague.—Would not that tend to shut the water away from the rivet?

Col. Stevens.—No.

Mr. Hague—The theory of expansion which the professor suggested was that the pressure on the inside of the blank ring would alternatively punch the head on one side and the other.

Mr. Stevenson Taylor.—I would like to say that Harlan & Hollingsworth Co., of Wilmington, have built a duplicate of this engine and these boilers. They say they have had no such trouble with their furnaces, although they have been used with forced draught. They make the ring bear on the bottom as was suggested, and the ring is not solid. I would like to ask if this ring is solid?

Mr. H. B. Roelker.—The ring is solid, yes, sir.

Mr. Taylor.—I would like to call attention to a point in reference to the engine. Col. Stevens spoke of the relative cost of the engines of the *Orange* and *Bergen*, but he did not take into consideration the cost of the *Bergen's* machinery to the *builders.* I am credibly informed that the machinery of the *Bergen* cost the builders, say 50 per cent. more than the contract price.

Mr. H. B. Roelker.—I beg to remind the member that the Delamater Iron Works burnt up while the engine was on the floor, and part of it had to be built twice.

Mr. Emery.—Troubles occur with boilers of this class under circumstances which at times it seems impossible to explain. I have built the boilers of eight or ten steamers from exactly the same specifications, and know that their construction was exactly the same, so far as it was possible to ascertain; and yet on one of the steamers, where the furnaces were built of iron instead of steel, the sheets cracked at the angles where they were flanged

outward and joined together through calking rings, not only in several of the sections of one furnace, but similarly in the other furnaces. The tests of the material were at the time satisfactory, and no explanation of the result could be obtained. From the fact that the other vessels had had no such difficulty, it was simply decided to put in another set of furnaces, using due care to test the material and watch its manipulation, and hope for better results. This work is now in progress.

Mr. J. F. Holloway.—The method of bracing or staying the boiler furnace as shown is quite different from our practice in the West. In regard to the method shown, I would offer an opinion as to the cause of the trouble, and also my idea of a method of overcoming the trouble.

The usual method of staying furnaces in locomotive boilers, as well as the side and crown-sheets of marine boilers in use on the lakes, is by the use of screwed stay bolts. These stay bolts are screwed in through the outer plates into and through the inner plates, and then both ends are riveted over cold. By this arrangement no strain is put upon the stay bolt or rivet. Now, in the arrangement as shown, you have a very long rivet; this rivet is driven hot, and when it cools off there is a severe strain put on the head of the rivet, and it does not, in my judgment, require any considerable amount of heating and cooling to increase the strain sufficiently to tear off the inner or riveted-over head. I know it is the practice in the East, on boilers which only carry low pressures, to use long bolts with thimbles between the plates, but it is new to me that this kind of construction is used on boilers intended to carry high pressures. All the marine boilers of the olden time carried low steam pressures, and the stay bolts were usually long bolts with heads and nuts. I think in this case, if the stay bolts had been screw bolts, put in cold and riveted over, there would have been no internal strains on them to assist in breaking the heads off.

Speaking of the strains which come to machinery, and castings of various kinds, few of us can imagine to what extent they exist by reasons of shrinkage, or by expansion due to heating and cooling. I never see a complicated and crooked casting but I think of it as being like a man who has the rheumatism; it is aching and straining in every direction.

I think in this case, very likely, the great length of the stay rivet, which is put in place hot, then riveted over, is, by cooling

off, strained to a point at which the head is quite ready to come off with slight additional strains.

Prof. J. B. Webb.—I remember a paper, which I believe was read before the American Association for the Advancement of Science some few years ago, where the writer had dealt with locomotive fire-boxes, and found the same action due to the expansion of the inner sheet of the water leg, which tended to break off the stay bolts. I believe he found that the expansion was the real cause of their failure. Now, as to the remedy in the present case, it is doubtful whether those horizontal rivets are of any advantage whatever. If the fire-box is hotter than the water when the fire is in it, then it must be expanded more than the

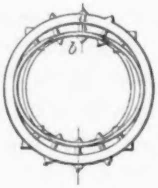


FIG. 88.

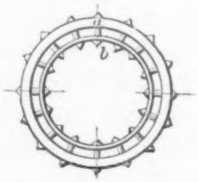


FIG. 89.

rings which are surrounded by the water. The heat is going through the metal from the fire to the water, so, of course, the fire-box is hotter than the ring. If the fire-box were riveted to the ring by its upper and lower quadrants only, leaving the side quadrants free, the increased expansion of the former would cause it to assume an elliptical form with the longest diameter horizontal, supposing it to be, of course, concentric with the ring when cold. Figs. 88 and 89 show such a construction; *a* is the ring, *b* the fire-box; the dotted lines indicating its circular form when cold, and the full lines its elliptical form when in use. If it be assumed, for example, that the fire-box is 36 inches diameter, and that its temperature averages 200 degrees higher than that of the ring *a*, then the horizontal diameter will be greater than the vertical by more than $\frac{1}{16}$ of an inch. As the steam acts to compress it in the direction of its least diameter, it follows that the rivets will act to hold the upper and lower quadrants from being forced inwards, and that no rivets are needed in the right and left quadrants. Fig. 89 illustrates the effect of rivets all round. Supposing the upper half of the fire-box to average 200 degrees hotter than the lower half,

which is protected considerably by ashes, the expansion of the upper half will exceed that of the lower by about $\frac{1}{50}$ of an inch at each side; so that while the top and bottom rivets would remain vertical, the horizontal rivets would have their inner ends forced downwards, and their centre lines would therefore be inclined from the horizontal as indicated by the dotted lines. The inner ends of the rivets would be forced downward $\frac{1}{50}$ of an inch, and a repetition of this a sufficient number of times might break off the heads of the rivets.

Mr. C. E. Hyde.—I would like to ask whether this accident might not have been caused by overheating on account of grease in the furnace?

Prof. Denton.—I understand that the furnace was fairly clean.

Mr. Hyde.—I had some experience a few years ago which goes to show that that thing may be caused by a very small quantity of grease which ordinarily would not be noticed. Unless that has been examined pretty closely, I should say that there was a possibility that that might have caused it.

Prof. Wood.—It is a wonder that the boiler stands at all. It is easier to explain why they fail under the severe stresses due to expansion and contraction. I was about to relate the same incident which Professor Webb referred to. I think many of us can confirm that action in regard to stay bolts. Now, if in this particular boiler the rim of iron is compressed all about equally, it would then retain a circular form, so that we have only to consider the fact of the extension of the upper part compared with the lower. As the upper part expands, it will tend to throw the upper part upward, elongating the vertical diameter and tending to shorten the horizontal one, thus tending to pull the heads off the bolts. The alternate, unequal heating and cooling of the shell will tend to shear the bolts and thus weaken them, rendering them more susceptible to the pulling action when the horizontal diameter shortens, until by repeated actions of this kind they ultimately fail. The remedy in such a case would be to make the bolts stronger.

*Mr. H. de B. Parsons.**—I preface my remarks by stating that the following theory of the failure of the rivets in the flues of the boiler of the *Bergen* is given with some reserve; for I have not seen the flues, nor have I examined the rivets, but advance

* Contributed after adjournment.

this theory on the facts as presented at the meeting, or at least as I understood them.

The stiffening ring, being so much deeper than the thickness of the flue, offers very much greater resistance to distortion than the flue; and also, being at a much more uniform temperature, it practically retains its circular form. On account of the position of the fire, the portion of the flue above the grate will be subjected to a greater degree of heat than the part below, and

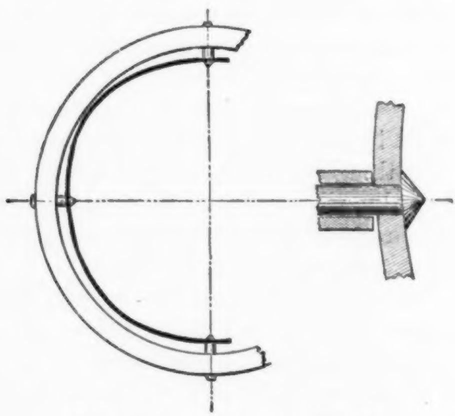


FIG. 90. FIG. 91.

in consequence the upper half will be more expanded than the lower. At sections of the flue between the stiffening rings, the flue will assume a smooth, although not a circular section. The rivets being held firm by the great rigidity of the stiffening ring, will cause the flue at that part to assume some such form, as is shown in an exaggerated manner in the illustration, Figs. 90 and 91.

The vertical plane through the top and bottom rivets would be a plane of symmetry, since from the nature of the case the disturbing elements to the right and left balance one another; and the top and bottom rivets, if good in the beginning, ought to show no signs of weakness.

On the other hand, the side rivets are the ones which produce the distortion; and since the rivets were riveted up tight when the metal was cold, any "play" which may take place will tend not only to shear the rivets, but also to throw off their heads. This play is caused by the side rivet, which prevents the upper

quadrant from expanding downward, and compels it to bulge outward. And since the upper quadrant bulges outward more than the lower, the lower quadrant will be thrown inward at the side rivet; or, in other words, the sheet of the flue will turn about the upper edge of the ferrule, as is shown in Fig. 91.

Should the rivets be made very large in comparison to the thickness of the flue, their extra strength might, and probably would, save them, but it would in no way remove the primal cause of weakness.

*Prof. J. E. Denton.** —I reply first to the questions as to how far excessive clearance, and loss of pressure during admission, are responsible for the failure of the *Bergen* to show much better economy than double-expansion engines such as are referred to by Mr. Emery.

A clearance of 14 per cent. in the high cylinder wastes only as much steam as would 2 per cent. of clearance in a single cylinder engine of equal ratio of expansion. Marine architects have, therefore, been very lavish of clearance in applying the piston valve, which naturally requires more clearance than slide valves. Engines are in use with piston valves, however, with as little as 7 per cent. clearance.

The clearance in the second and low cylinders do not directly increase steam consumption, but there is an indirect increase by the loss of area between the indicator cards. The lowering of the initial pressure in the high cylinder lowers the ratio of expansion for a given horse-power, and thereby prevents the highest economy. Except in the matter of this low initial pressure, the *Bergen* is not inferior to any of the recent triple-expansion engines, as the clearances are about as great in them as in the *Bergen*.

Allowing for differences in initial pressure, ratio of expansion, cushion, and clearance, between the SS. *Meteor* and the *Bergen*, the latter should have used about 17.8 pounds of steam instead of 18.3, taking the *Meteor* to have used 15 pounds. In other words, the low economy of the *Bergen* is practically ascribable to low ratio of expansion, due to low initial pressure. This point is further discussed in the body of the paper under the heading, "Combined Card of *Bergen*."

Regarding the various opinions on the failure of the furnace,

* Author's Closure, under the Rules.

with two exceptions they appear to support the idea that strains could easily occur in the ring rivets, which would render the structure defective to a degree corresponding with the weakness described. I agree with the views expressed by Mr. Holloway and Mr. Emery as to the unreliability of riveted work under these conditions, and in general with the theories of expansion and contraction enunciated by the other speakers. The two exceptions are the idea of grease being responsible, and the opinion that making the ring eccentric to the furnace, or touch the furnace directly for a portion of its circumference, would eliminate the possibilities of such trouble as experienced with the *Bergen*. Greasy deposits on the furnace unquestionably could cause leakage, but when the furnaces were removed the writer examined the iron carefully and found no traces of a grease deposit. As to the eccentric ring arrangement, the *City of Kingston* had its furnaces so arranged, and experienced the same troubles as the *Bergen*, until the rivets were *tapped through the boiler-plate*, as suggested by Mr. Holloway.

At the date of this writing (Feb. 5, 1890) the *Bergen* has been fitted with corrugated furnaces, and has been in regular daily service several weeks without any sign of defects in the furnaces; but Craig's circulators are in use to circulate the water under the furnace.

CCLXXVI.

THEORY AND DESIGN OF CHIMNEYS; WITH CRITICISMS ON THE COMMON THEORY, AND EXPERIMENTAL DATA.

BY HORACE B. GALE, ST. LOUIS, MO.

(Member of the Society.)

CHIMNEYS are usually built to serve two purposes—to create a draught for supplying air to the fire, and to discharge the smoke and gases high enough above the ground to make them comparatively inoffensive. The employment of forced draught is so rapidly extending, that the use of chimneys for the first-mentioned purpose in large establishments may some time become a thing of the past; but it is likely that they will be always needed for the second purpose; and in cases where the present ordinary methods of burning coal under boilers continue to be employed, it will probably be found better—for reasons given hereafter—to use a combination of chimney draught and forced draught, rather than either alone. It appears, therefore, that the problem of determining the best forms, dimensions, and materials for chimneys to serve any desired purpose will continue to be an important matter to the engineer.

OBJECTIONS TO THE COMMON THEORY.

It has long seemed to the writer that the theory of chimney draught promulgated by Peclet, Rankine, Morin, and Weisbach, and generally accepted by more modern writers, contains certain errors, on which have been based conclusions and formulæ which are not supported by the results obtained in practice. The disagreement has, indeed, led some engineers to discard the theory and to depend upon rule of thumb.

As an illustration, there is derived from the common theory a formula for the number of pounds of fuel which may be burned per hour with a given chimney, which is usually expressed by an equation of the form $F = CA\sqrt{H}$; in which H is the height of

the chimney, A the area of its cross section, and C a coefficient whose value is variously stated, ranging usually from 10 to 15. Some writers state this formula as if it were a matter of indifference, in regard to the draught, whether—other things remaining the same—we build a chimney 81 feet high and 2 feet square inside, or 16 feet high and 3 feet square inside, the product $A\sqrt{H}$ being the same in each case. The 16-foot chimney would be considerably cheaper, and the main reason why it is not preferred is that practical engineers know that it is not an equivalent for the 81-foot chimney. Other writers qualify the rule by the statement that it applies only to cases in which the total grate area is about eight times the area of the chimney. It is, however, sometimes desirable to make the grate area more, or less, than eight times the chimney area.

For example, by this rule a 54-inch, round, iron stack, 64 feet high, having a sectional area of 16 square feet, will be just able to burn $15 \times 16 \times \sqrt{64}$ (= 1920) pounds of coal per hour (using the maximum value of the coefficient). Making the total grate area eight times the section of the chimney would give us 128 square feet of grate; or, we may say that the chimney would just suffice for four boilers, each having 32 square feet of grate surface, and burning 15 pounds of coal per hour per square foot. If at some future time it is desired to add two more boilers of the same size, the question arises whether or not they may be connected to the same stack; or, in case enlarged chimney capacity is necessary, whether it will be more economical to build an additional stack, or to increase the height of the old one. To the latter question the ordinary chimney theory gives no satisfactory answer. The formula indicates that the present stack is doing all the work of which it is capable; but if the experiment is made of connecting to it the two additional boilers, it is found to answer about as well for six as it did for four; and the amount of coal burned per hour is increased nearly in proportion to the increase in grate surface. If, on the other hand, the number of boilers connected to the stack is reduced, the amount of coal burned per hour is found to be reduced in nearly the same ratio.

Within certain limits, the total fuel consumption is almost independent of the area of the chimney, and depends mainly upon the grate area; but this cannot be taken as a general law, any more than the other; for if we continue to connect more boilers to the same stack, we soon reach a point where the coal con-

sumed ceases to increase in proportion to the grate surface. In no case, however, does the quantity of coal burned on a given grate vary in the same ratio as the area of the chimney. These are facts well substantiated by experience.

Again, from the same theory of draught is deduced a law, so called, which asserts that no matter what the dimensions of the chimney and furnace, the maximum draught will be attained when the temperature of the chimney gases is such that their density is just one-half the density of the outside air. This would make the temperature of maximum draught in any chimney about 600° Fahrenheit. Experiments upon chimneys of ordinary proportion show, nevertheless, that the draught is stronger, and that more coal is burned per square foot of grate, when the temperature rises above 600°.

Had not the writer thoroughly satisfied himself of the incorrectness of the common theory upon these points by experiment, as well as by mathematical analysis, his high respect for the authority of Rankine would deter him from offering as a substitute the theory herein set forth, which appears to him to be in better accord with the truth.

GENERAL PRINCIPLES OF CHIMNEY DRAUGHT.

If one end of a glass *U* tube, about half full of water, is connected with the space in the bottom of a chimney, and the other end left open to the air, it is found that when the chimney is cold the water stands at the same level in the two branches of the tube; but when there is a fire in the furnace, and the chimney is full of hot gas, the level of the water in the branch of the *U* connected to the chimney is slightly higher than that in the other branch, showing that the pressure in the bottom of the chimney is less than the external atmospheric pressure by an amount corresponding to the difference in level. One inch difference in level would correspond to a reduction of pressure of 5.2 pounds on the square foot; but the difference is more frequently less than this, and is usually stated in tenths of an inch. This reduction of pressure in the bottom of the chimney is explained by the fact that the column of gas in the chimney, being expanded by heat, is not as heavy as a similar column of the outside air.

Now, it is the excess of the external pressure over the pressure in the bottom of the chimney which causes the current of air to

flow through the furnace into the chimney; and upon the quantity of air forced through the grate depends the quantity of fuel which can be burned. It follows that, with a given furnace, the chimney which will enable the most coal to be burned in a given time is that which has the smallest internal pressure at the bottom. Let us first ascertain, therefore, on what the reduction of pressure in the bottom of a chimney depends.

If the column of hot gas in the chimney were standing still, the excess of the external over the internal pressure at the bottom, expressed in pounds on the square foot, would be equal to the excess of the weight of a column of the outside air as high as the chimney, and one foot square, over the weight of a similar column of the heated gases in the chimney. But as the gases are in motion, a certain back pressure is generated by their friction on the sides of the chimney, and this back pressure must be subtracted from the difference in weight between the external and internal columns, in order to find the real reduction of pressure at the bottom of the chimney; that is, in order to find the effective pressure acting to force air through the furnace.

Let P = the excess of the external over the internal pressure at the base of the chimney, in pounds on the square foot;

H = the height in feet of the top of the chimney above the fire grate;

D_a = the density, in pounds avoirdupois per cubic foot, of the outside air;

D_s = the mean density of the gases in the stack. Then the weight of a column of gas one foot square in the chimney is HD_s, and the weight of a similar column of the outside air is HD_a.

Now, the effective pressure per square foot acting to produce the draught through the furnace is equal to the difference between the weights of these columns, minus the back pressure due to friction in the chimney. Let P_f represent this back pressure in pounds per square foot; then we have

$$P = H(D_a - D_s) - P_f. \quad . \quad . \quad . \quad . \quad (1)$$

The density of a gas is inversely as its absolute temperature; and if we let T_a = the absolute temperature of the outside air,

and T_s = the mean absolute temperature of the gases in the stack, we may substitute $\frac{39}{T_a}$ for D_a, and $\frac{40}{T_s}$ for D_s, the density of the chimney gases exceeding that of air at the same temperature approximately in the ratio 40 to 39. These constants are adapted to an elevation of about 340 feet above the sea level. Making these substitutions in the last equation gives us

$$P = H\left(\frac{39}{T_a} - \frac{40}{T_s}\right) - P_f. \quad . \quad . \quad . \quad . \quad (2)$$

Let V = the mean velocity of the gases in the chimney,
M = the inner perimeter of the stack, and
A = its sectional area.

Now, the total force required to overcome the friction in the stack is proportional to its interior surface—that is, its perimeter multiplied by its height—to the density of the chimney gases (or inversely to their temperature), and directly to the square of their velocity; and to find the back pressure per square foot we must divide the total force by the sectional area of the chimney in square feet. Therefore we may write

$$P_f = C\frac{MHV^2}{T_sA}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which C is a coefficient to be determined by experiment, and depending for its value upon the nature of the surface.

In this formula, V, the velocity of the chimney gases, is equal to the volume discharged per second, divided by the area of the cross section; and the volume is equal to the weight, divided by the density. Therefore, if we let W = the weight of gas discharged per second, we have $V = \frac{W}{D_sA}$; or, as $D_s = \frac{40}{T_s}$, $V = \frac{T_sW}{40A}$. Substituting this value of V in Equation 3, gives, for the back pressure due to friction in the stack,

$$P_f = \frac{CMHT_sW^2}{1600A^3}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

EXPERIMENTAL DETERMINATION OF COEFFICIENT OF FRICTION.

The author has made a few experiments to determine the value of the coefficient C for brick and iron stacks. The means at hand were not such as to permit of very accurate determinations, and the number of experiments thus far made is so small that the

values here given can be considered only as rough approximations.

The method employed was to take the temperature of the gases entering the chimney by a mercury thermometer, and to measure the effective pressure at the bottom of the chimney by a draught gauge, such as has been described. The pressure P_f, due to friction in the chimney, was determined by noticing the increase in effective pressure, caused by momentarily closing a damper in the flue between the furnace and chimney. By this means the motion of the gases was stopped, and the draught gauge measured the difference in weight between the outer and inner columns. By closing the damper, P_f in Equation 2 is reduced to zero, and we have $P = H\left(\frac{39}{T_a} - \frac{40}{T_s}\right)$. The value of T_a, the temperature of the outer air, being known, the equation then gives us the value of T^s, or the mean temperature of the chimney gases. The difference between this temperature and the observed temperature at the bottom measures the amount of cooling in the stack.

The weight of gas discharged per second, W, was determined in the following way: Observations were made at regular intervals during a ten-hour boiler trial, for which time all the conditions were kept as uniform as possible. The total weight of coal consumed in that time was carefully determined, and samples of the coal and of the flue-gases were taken at intervals for analysis. A comparison of the analyses of the coal and of the flue-gases enables one to calculate the weight of the gas delivered from the known weight of the fuel consumed. The dimensions of the chimney being known, C becomes the only unknown quantity in Equation 4, and its value may be calculated. The measurements show that in ordinary cases the back pressure due to friction in the stack is a relatively small quantity, in boiler chimneys seldom rising as high as one-fourth of the effective pressure.

From the experiments thus far made are deduced the following provisional values of the constant C in Equations 3 and 4:

For an iron stack, $C = .012$.
For a brick stack, $C = .016$.

DESIGN OF CHIMNEY TO DO GIVEN WORK.

The problem of designing a chimney to do a given work may be divided into two parts: first, that of ascertaining the draught pressure necessary to burn the desired quantity of fuel in the furnace;

second, that of determining the dimensions of a chimney which will produce the required draught at the least expense.

The number of pounds of coal which can be burned per second on a given grate is equal to the weight of air forced through, divided by the number of pounds of air required for the combustion of a pound of coal under the given conditions. The weight of air forced through per second is equal to the area through which it is admitted, multiplied by its velocity, and by its density, or weight per cubic foot.

Let a = the total area of the openings through which air is admitted to the fire. In an ordinary furnace this would be the area of the openings between the grate-bars, plus the area of any orifices for the admission of air above the fire.

Let v = the mean velocity of the current entering through the space a.

Let B = the number of pounds of air supplied per pound of fuel. Then the number of pounds of fuel burned per second is $\frac{avD_a}{B}, = \frac{39av}{T_aB}$, and the number of pounds burned per hour, $F, = \frac{3600 \times 39av}{T_aB}$; or,

$$v = \frac{T_aBF}{140400a}. \qquad (5)$$

Determination of Resistance of Furnace.

Now, the velocity v with which the air enters between the grate-bars depends upon the effective pressure P acting between the space beneath the grate-bars and the bottom of the chimney, and also upon the resistance encountered by the gases between these two points. The pressure P is expended in various ways. A part of it may be considered to be employed to impart the velocity v to the entering air; part is consumed in overcoming the frictional and other resistances offered to the passage of the gases through the bed of coals, the boiler-tubes, and other portions of the furnace and flues; and part is used in imparting to the gases the final velocity V with which they traverse the chimney. Some measurements, made by the writer, of the pressures expended in these various ways in the case of a stationary boiler furnace of ordinary construction, gave the following results:

Area of grate, 22.5 sq. ft.
Area of grate opening, 6.19 sq. ft.

Area for admitting air above fire, 0.45 sq. ft.
Total admission area, a, 6.64 sq. ft.
Tube calorimeter of boiler (36 tubes, 4″ x 16′), 2.74 sq. ft.
Coal burned per sq. ft. of grate per hour, 13.5 lbs.
Lbs. of air supplied per lb. of coal (from analyses), 21 lbs.
Temperature of boiler room, 60° F.
Outside temperature, 40° F.
Temperature of chimney gases, 460° F.
Height of top of chimney above grate, 72 ft.
Area of chimney (round iron stack), 4 sq. ft.
Length of horizontal iron smoke-flue, 24 ft.
Area of cross section of flue, 4 sq. ft.

Pressures in lbs on the sq. ft.—

Required to produce entrance velocity (3.6 ft. per sec.)	.013
Required to overcome resistance of fire-grate	.91
Required to overcome resistance of combustion chamber and boiler-tubes	1.23
Required to overcome resistance in horizontal flue	.06
Required to produce discharge velocity (11.2 ft. per sec.)	.085
Total effective draught pressure, P	2.298
Back pressure due to friction in stack	.19
Total static pressure produced by chimney ($H[D_a - D_s]$)	2.488

It is evident that the greater part of the draught pressure is required to overcome the resistance offered to the passage of the gases in the fire-grate and boiler-tubes.

The frictional resistance encountered in any part of the furnace is proportional to the density of the gas *at that point*, multiplied by the square of its velocity at that point; but, *for a given furnace*, the velocities and the densities in the different parts bear an approximately constant relation to each other; therefore the entire frictional resistance for any furnace may, with sufficient accuracy, be made proportional to the square of the entrance velocity v, and may be represented by Kv^2, in which expression K is a coefficient which would have different values for different types of furnace.

The velocity of the gas *in the chimney*, however, bears no necessary relation to its velocity in the fire-grate and tubes, and indeed may be made to assume any desired ratio to it by varying the relative areas of the passages. It is therefore *not* allowable to assume that the resistance to the passage of the gas through any given furnace may be found by multiplying the square of the *chimney velocity* by a constant. This very apparent error is nev-

ertheless made by most writers on the subject of chimneys, and is the foundation of the peculiar formulæ before referred to.

In order to determine the proper dimensions for a chimney, we must know the value of the coefficient of resistance for the furnace, K. Its value would evidently depend principally upon the thickness and compactness of the bed of coals, and upon the amount of contraction of area in the boiler-tubes. The value of K in any given case can be determined with accuracy only by experiment on a furnace precisely like the one we are to use; but as this method is usually impracticable, the following approximate method of calculation may be employed. Let us assume the effective pressure P to be applied to the following objects, viz.: to imparting the entrance velocity v, to overcoming the resistances offered successively by the fire-grate, the combustion chamber, the boiler-tubes, and the smoke flue, and to imparting to the gases the final discharge velocity V.

Let r be a coefficient of resistance for the fire-grate and bed of coals;
c a similar coefficient for the combustion chamber;
b a coefficient for the boiler-tubes;
f a coefficient for the flue leading to the chimney;
l the length of this flue in feet;
d its diameter in feet;
a'' the area of cross section of the flue in sq. ft.;
a' the tube calorimeter in sq. ft.;

then $$K = \frac{D_a}{2g} + r + c + b\left(\frac{a}{a'}\right)^2 + f\frac{l}{d}\left(\frac{a}{a''}\right)^2 + \frac{D_s}{2g}\left(\frac{V}{v}\right)^2. \qquad (6)$$

From the measurements already given are derived the following provisional values for the constants in this equation:

$$\frac{D_a}{2g} = .0011, r = .07, c = .001, b = .016, f = .00016, \frac{D_s}{2g} = \frac{0.6}{T_s}.$$

There is need of more extended and precise experiments than the writer has yet found time to make, in order to determine the values of these coefficients with accuracy. Substituting these values in the equation, and taking values for the remaining quantities from the example cited, we have $K = 0.18$. For ordinary boiler furnaces, we may take 0.2 as a fairly safe value for K; but, as its value varies greatly in different cases, it is better to compute t for each special case by the method just described.

If in the expression $P = Kv^2$, we substitute for v its value from Equation 5, we obtain,

$$P = K\left(\frac{T_a BF}{140400\,a}\right)^2. \quad . \quad . \quad . \quad . \quad . \quad (7)$$

This equation gives us the reduction of pressure in pounds on the square foot needed at the bottom of a chimney which is to burn F pounds of fuel per hour, in a furnace having an area for air admission a (in square feet) and a coefficient of resistance K, allowing B pounds of air per pound of fuel.

In order to find the height of the chimney necessary to produce the required draught pressure, we have, by combining Equations 1 and 4,

$$P = H\left(D_a - D_s - \frac{CMT_s W^2}{1600 A^3}\right). \quad . \quad . \quad . \quad . \quad (8)$$

In this equation, W is the weight of the chimney-gas discharged per second, which is usually about five per cent. greater than the weight of the air supplied in the same time. Therefore we may put

$$W = \frac{1.05\,BF}{3600}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and substituting this value in Equation 8, we have, after combining the numerical terms,

$$P = H\left(D_a - D_s - \frac{CMT_s B^2 F^2}{18{,}808{,}000{,}000.\ A^3}\right). \quad . \quad . \quad . \quad (10)$$

By placing the two values of P from Equations 7 and 10 equal to each other, we obtain,

$$H = \frac{K}{D_a - D_s - \dfrac{CMT_s B^2 F^2}{18{,}808{,}000{,}000.\ A^3}} \left(\frac{T_a BF}{140{,}400.\ a}\right)^2. \quad . \quad . \quad (11)$$

This equation enables us to calculate the height of a chimney required to do a given work, when its cross section is known.

In the case of a round chimney, the inside perimeter, M, is equal to 3.54 $A^{\frac{1}{2}}$; and for any other shape, we may put $M = 3.54\ A^{\frac{1}{2}}R$, where R is the ratio of the actual perimeter to the circumference of a circle having the same area. R is therefore a factor

depending on the *form* of the cross section. For a square chimney, $R = \frac{2}{\sqrt{\pi}} = 1.13$; for a round chimney, its value is unity. Substituting for M in Equation 11 the value just deduced, we have,

$$H = \frac{K}{D_a - D_s - \frac{CRT_sB^2F^2}{5,300,000,000.\ A^{\frac{5}{2}}}} \left(\frac{T_aBF}{140400a}\right)^2. \quad . \quad . \quad (12)$$

It is evident from an inspection of this equation that by assuming different values for A, the sectional area, we may obtain an indefinite number of chimneys of different heights, any one of which will be capable of doing the work required. The best value to assign to A is usually that which will make *the cost of the chimney a minimum.*

MOST ECONOMICAL PROPORTIONS FOR CHIMNEY.

Let E = the cost of the chimney; then, to find the most economical value for A, we must express E as a function of A, and put the differential coefficient $\frac{dE}{dA}$ equal to zero. A comparison of the cost of several chimneys has led to the conclusion that, in so far as it depends upon its dimensions, the cost of a chimney is nearly proportional to the $\frac{1}{3}$ power of the area, multiplied by the $\frac{3}{2}$ power of the height. Without fixing the precise value of the exponents, we may write, as a general expression,

$$E = C''A^mH^n, \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

in which C'' is a constant. Substituting in this equation the value of H from Equation 12, and placing the differential coefficient, $\frac{dE}{dA}$, equal to zero, we obtain, after some reduction,

$$A^{\frac{5}{2}} = \frac{(m + \frac{5}{2}n)\ CRT_sB^2F^2}{5,300,000,000.\ m\ (D_a - D_s)}. \quad . \quad . \quad . \quad (14)$$

This equation determines the value of A, which will make the cost of the chimney a minimum; and in order to find the corresponding height, we may substitute this value in Equation 12, which gives us

$$H = \frac{(n + \frac{2}{5}m)\ K}{n\ (D_a - D_s)} \left(\frac{T_aBF}{140,400.\ a}\right)^2. \quad . \quad . \quad . \quad (15)$$

In order to simplify the foregoing equations, and render them more convenient for use, we may assume an average value for the temperature of the air, say 59° Fahrenheit. This makes the absolute temperature, $T_a = 59 + 461 = 520$. Therefore $D_a = \frac{39}{520} = .075$; and we may also put $D_s = \frac{40}{T_s}$; whence $D_a - D_s = \frac{.075\,(T_s - 533)}{T_s}$. Making these substitutions in Equation 11, and reducing, gives, as a general equation for determining the height of a chimney of known cross section, which is to do a given work,

$$H = \frac{KT_s}{T_s - 533 - \frac{CM}{A^3}\left(\frac{8T_sBF}{300{,}000}\right)^2}\left(\frac{BF}{74a}\right)^2. \quad . \quad . \quad (16)$$

In this equation,

H = the height in feet of the top of the chimney above the grate;

K = the coefficient of resistance for the furnace;

T_s = the mean temperature of the gas in the stack in Fahrenheit degrees above the absolute zero;

C = the coefficient of friction of the gas on the sides of the stack (= .012 for iron, .016 for brick);

M = the inside perimeter of the stack in feet;

A = the area of its cross section, in square feet;

B = the number of pounds of air supplied to burn a pound of fuel;

a = the total area for admission of air, in square feet;

F = the number of pounds of fuel to be burned per hour.

The formula may be still further simplified by substituting average values for C and B. Assuming $C = .014$, and $B = 21$, gives

$$H = \frac{KT_s}{T_s - 533 - \frac{M}{A^3}\left(\frac{T_sF}{15{,}000}\right)^2}\left(\frac{F}{3.5a}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

In Equation 14 the same substitutions may be made, and to m and n may be assigned the values $\frac{1}{3}$ and $\frac{2}{3}$ respectively. In this equation also the quantity $\frac{T_s}{D_a - D_s}$ is so nearly constant for the temperatures ordinarily found in chimneys, say from 200 to 2,000 degrees on the common Fahrenheit scale, that its possible range of variation cannot seriously affect the value of A. We may there-

fore put in place of it its average value, about 36,700, which gives, as the most economical area for a chimney which is to burn F pounds of coal per hour,

$$A = .05R^{\frac{3}{4}}F^{\frac{3}{4}}\,; \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

which is represented with sufficient accuracy for the purpose by the more convenient formula,

$$A = .07R^{\frac{3}{4}}F^{\frac{3}{4}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

an equation which is easily solved without the use of logarithms. R, it will be remembered, is a factor depending upon the form of the cross section, being the ratio of the perimeter of the chimney to the circumference of a circle having the same area. For a circular stack, $R = 1$, which makes the formula for the area of a circular stack which is to burn F pounds of coal per hour,

$$A = .07F^{\frac{3}{4}}\,. \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

For a square chimney, $R = 1.13$, which would make the coefficient about 6 per cent. greater, but this difference is unimportant, and Equation 20 may be applied to either round or square chimneys.

We have now deduced the useful law that in order to attain the greatest economy, the area of cross section of a chimney should be proportioned approximately to the $\frac{3}{4}$ power of the weight of fuel that is to be burned by means of it. It should be noticed that extreme exactness, if it were possible, would not be important here, as even considerable variations in cross section on either side of the best value have a comparatively slight effect upon the economy. Indeed, for chimneys of ordinary proportions, a sufficient approach to the most economical value can usually be attained by the handy rule that *the sectional area of the chimney in square feet should be equal to the number of pounds of fuel to be burned per minute.* This would make the velocity of the gases in the chimney usually between seven and eleven feet per second. For unusually large chimneys the more exact formula of Equation 20 is recommended.

If the area of the chimney is determined by the foregoing methods, the corresponding formula of Equation 15 may be used to determine the height. This equation also may be simplified for use by substituting the same values as before for T_a, n, m, and

$(D_a - D_s)$, which gives, for the most economical height of a chimney to burn F pounds of fuel per hour,

$$H = \frac{KT_s}{T_s - 533}\left(\frac{BF}{70a}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

For those cases in which the temperature of the chimney gases is between the limits of 150 and 600 degrees Fahrenheit, which would include most boiler chimneys, the quantity $\frac{T_s}{T_s - 533}$ is approximately equal to $\frac{1111}{t}$, where *t is the temperature reckoned from the ordinary zero of Fahrenheit's thermometer*. For such cases, therefore, we may reduce the formula to a still simpler form by substituting this value, and assuming B, as before, equal to 21; which gives, as a rule for determining the most economical height of boiler chimneys,

$$H = 100\,\frac{K}{t}\left(\frac{F}{a}\right)^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

In ordinary boiler furnaces the area for admission of air to the fire, a, may be considered roughly as $\frac{1}{3}$ of the grate area, and when the ratio of grate surface to tube calorimeter is between 6 and 9, K may be taken approximately as 0.2. Let G = the area of the grate in square feet; then, on the assumptions just made, we have, approximately,

$$H = \frac{180}{t}\left(\frac{F}{G}\right)^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

This formula makes the height of a boiler chimney proportional to the square of the rate of combustion per square foot of grate, divided by the common temperature of the chimney gas. It can be considered, however, only as an extremely rough approximation, and is not recommended for general use.

It is apparent, from the foregoing discussion, that the rules which are sometimes given, making the height of the chimney about three-tenths of the square of the rate of combustion, without regard to the temperature of the escaping gases, can hardly claim the dignity even of approximations. If such a rule gave just sufficient height to do the required work with a chimney temperature of 600 degrees Fahrenheit, it would give only one-half the proper height for a temperature of 300 degrees.

Equation 23 is mainly useful as a rough guide in deciding upon the rate of combustion to be employed. As K varies very considerably with different types and proportions of furnaces and boilers, it is best, in designing a chimney, to use Equation 22, and substitute the proper value of K, as nearly as it can be ascertained.

By a simple transformation of Equation 22, we can calculate also the number of pounds of fuel that can be burned per hour in a given furnace with a chimney of height H, *provided its sectional area is made sufficient*; viz.,

$$F = \frac{a}{10}\sqrt{\frac{Ht}{K}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Again, by making

$$a = 10\,F\sqrt{\frac{K}{Ht}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

we are enabled to calculate the area required for the admission of air, in order to burn F pounds of fuel per hour with a chimney of height H, and, knowing the area of opening per square foot of grate, we can determine the area of the grate surface needed.

In cases where the number of pounds of air supplied per pound of fuel differs much from the ordinary steam boiler practice, or where the chimney temperature is outside the limits of 150 and 600 degrees Fahrenheit, as is the case in some metallurgical works, we should go back to Equation 21, which takes account of these factors. It should be remembered, also, that Equation 21, and all those following which are derived from it, apply only to cases in which the sectional area of the chimney has been determined by the method herein described, or an equivalent one. If the sectional area is made *smaller* than its most economical value, the frictional resistance to the passage of the gases rapidly increases, and the effective draught is reduced, unless the height is increased sufficiently to counteract the diminished area. If, therefore, for any special reason it is desired to make the cross section of the chimney smaller than the value given by Equation 20, the general formula of Equation 16 should be used to determine the height. If, on the other hand, the area is made *larger* than the most economical value, although the frictional resistance encountered in the stack will be diminished, the margin for reduction is

too small to allow any considerable diminution in height on that account; therefore, one of the equations, 21 or 22, may be safely applied to all cases in which the chimney section is not too small.

AREA OF MAXIMUM DRAUGHT, FOR CHIMNEY OF GIVEN HEIGHT.

Enlarging the sectional area of the chimney, as it diminishes the velocity of the escaping gases, and therefore also the back pressure due to friction, will increase the effective draught, at least, within certain limits; but when the diameter is enlarged enough to bring the velocity of the gas down to about five feet per second, the loss due to friction in the stack becomes usually so small as to be unimportant. On the other hand, as we increase the diameter, and reduce the chimney velocity, we increase the heat-radiating surface of the chimney, and also give the gases more time to cool in going from the bottom to the top, so that we thereby lower the temperature in the chimney; and we shall eventually reach a point where the increase in weight of the gas, due to cooling, will more than offset the reduction of back pressure due to diminished friction. Any further increase in the sectional area of the chimney will then diminish, instead of augmenting, the draught. It follows that for a chimney of any given height, discharging a given weight of gas, which enters it at a known temperature, there will be a certain area which will produce a stronger draught than any other. In order to determine this area, it will be necessary to investigate the relation of the dimensions of the chimney to the mean temperature of the gases.

The fall of temperature of the gas, due to cooling in the stack, will be proportional to the quantity of heat radiated from the chimney in a unit of time, divided by the weight of gas discharged in the same time. The heat radiated from the chimney per second will be proportional roughly to its surface, or its height multiplied by its perimeter, and to the difference between the mean temperature of the gases in it and the outside temperature. That is to say, the loss of temperature sustained by the gases in going from the bottom to the top will be proportional to $\frac{HM(T_s - T_a)}{W}$. Let T_1 be the temperature of the gases entering the stack. Now, the mean temperature of the chimney will be less than the temperature at the bottom by a quantity which is nearly proportional to the amount of cooling; and, remembering

that M in the last expression is proportional to $RA^{\frac{1}{2}}$, we may write the equation

$$T_s = T_1 - C' \frac{HRA^{\frac{1}{2}}(T_s - T_a)}{W}, \quad . \quad . \quad . \quad . \quad (26)$$

in which C' is an experimental coefficient.

The experiments already referred to give, as a provisional value of C' for an iron stack, $C' = .002$; for a brick stack, or a lined iron stack, the amount of cooling would be considerably less, and C' may be taken as about .0003. There is need of further experiment to determine these values accurately; however, as the loss by cooling in the chimney is usually comparatively small, a considerable variation in these constants would have but a small effect. In the case of brick chimneys, the leakage of cold air in through the brickwork often has a serious effect in lowering the temperature in the chimney, and interfering with the draught; and the leakage of cold air through cracks in the brick boiler setting has frequently a still more hurtful effect.

By transforming Equation 26, we obtain, for determining the mean temperature of the gases in the chimney, where T_1 is the temperature at the bottom:

$$T_s = \frac{WT_1 + RC'HT_aA^{\frac{1}{2}}}{W + RC'HA^{\frac{1}{2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Also, by combining Equations 2 and 4 and putting $M = 3.54\, RA^{\frac{1}{2}}$, we have for the effective draught pressure corresponding to a temperature T_s,

$$P = H\left(\frac{39}{T_a} - \frac{40}{T_s} - \frac{3.54\,CRW^2T_s}{1600A^{\frac{5}{2}}}\right). \quad . \quad . \quad . \quad . \quad (28)$$

In order to find the area which will give us the maximum draught in a chimney which is to discharge a given weight of gas per second, it is necessary to substitute for T_s, in Equation 28, its value from Equation 27. This expresses P as a function of A, and by placing the differential coefficient $\frac{dP}{dA}$ equal to zero, we obtain an equation which determines the value of A, that will make P, or the effective draught pressure, a maximum. The equation thus obtained can be only approximately solved. It gives,

as the approximate value of A for maximum draught in a stack of height H,

$$A' = .07 \frac{WT_1}{\sqrt[3]{\frac{C'}{C} H(T_1 - T_a)}}; \quad . \quad . \quad . \quad . \quad (29)$$

or, inserting the values of C' and C for an iron stack, and allowing 22 pounds of air per pound of fuel,

$$A' = .0008F \frac{T_1}{\sqrt[3]{H(T_1 - T_a)}}. \quad . \quad . \quad . \quad . \quad (30)$$

For a brick, or lined iron, stack, the same method gives

$$A' = .0016F \frac{T_1}{\sqrt[3]{H(T_1 - T_a)}}. \quad . \quad . \quad . \quad . \quad (31)$$

According to these equations, the sectional area for maximum draught in the case of a brick stack is about twice as great, or the velocity of the escaping gases one-half as great, as in an unlined iron stack receiving an equal weight of gas at the same temperature. As has been already mentioned, the values given to the numerical coefficients in these formulæ are not claimed to be exact, but the form of the expression is believed to be correct. The exact value of the area of maximum draught is not important, as there is a wide range on each side of this value for which the effective draught pressure will vary only a very slight amount, as may be seen by working out the value of P for areas ranging from one-half to three times that given by Equation 29.

The most economical cross section (Equation 20) is usually, as might be expected, smaller than the area of maximum draught; and bearing in mind the very slight diminution in the strength of the draught produced by exceeding even the latter area, it is evident that one need have usually no fear of impairing the draught by enlarging a chimney to three or four times the most economical cross section. If the enlargement is much greater than this, however, the draught begins to be seriously impaired; but usually the only objection to making a chimney too large is the unnecessary expense involved. It is probable also that when the velocity of the gases in the chimney is very slow, the draught is more liable to be interfered with by the wind, and thereby rendered unsteady.

This fault can often be remedied by a damper at the top of the chimney.

* [It should be noticed, also, as resulting from the principles developed in this section, that when the area of a chimney is large in proportion to its work, the draught, *at the time of starting the fire*, will not be as good as would be had from a smaller chimney. For the same reason, a chimney of small cross section in proportion to its work will *recover its power* somewhat more rapidly after the fire has been damped. The difference is more noticeable in iron chimneys than in brick ones, owing to the more rapid radiation of heat from the former.

When, therefore, it is especially desirable to be able to *start a fire quickly* by means of the draught of an iron stack, the area may with advantage be reduced somewhat below the most economical value, and a corresponding increase made in the height. For use in such a case, the numerical coefficients of equations 20 and 22 might be modified, for example, so as to make

$$A = .05\ F^{\frac{1}{2}}, \text{ and } H = 112\ \frac{K}{t}\left(\frac{F}{a}\right)^2.]$$

VELOCITY OF MAXIMUM DRAUGHT.

By combining Equations 27 and 28, as before, and placing the differential coefficient $\frac{dP}{dW}$ equal to zero, we may obtain an equation which determines the value of W, or the weight of gas discharged per second, which will make the draught pressure a maximum in a chimney of given dimensions. Performing this operation, and reducing, we have, as a close approximation,

$$W = 20\frac{A}{T_1}\sqrt[3]{\frac{C}{C}\,H\,(T_1 - T_a)}. \quad . \quad . \quad . \quad . \qquad (32)$$

If we let V_1 represent the velocity of the gases at the bottom of the chimney, the volume of gas delivered into the chimney per second will be $A\,V_1$, and its weight will be the volume multiplied by the density; or, as the density is equal to $\frac{40}{T_1}$, we may put $W = \frac{40\,A\,V_1}{T_1}$ and substituting this value in Equation 32, we

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* Added after adjournment.

obtain, as the velocity which the gases should have at the bottom of a chimney in order to produce the maximum draught,

$$V_1 = 0.5 \sqrt[3]{\frac{C'}{C} H (T_1 - T_a)}. \quad . \quad . \quad . \quad . \quad (33)$$

Inserting the values of C' and C for an iron stack, we obtain,

$$V_1 = .28 \sqrt[3]{H(T_1 - T_a)}; \quad . \quad . \quad . \quad . \quad . \quad (34)$$

and for a brick stack we have

$$V_1 = .14 \sqrt[3]{H(T_1 - T_a)}; \quad . \quad . \quad . \quad . \quad . \quad (35)$$

showing that the velocity needed to produce the maximum draught in an unlined iron stack is about twice as great as that required in a brick stack.

According to Equation 34, the velocity of maximum draught for iron stacks would ordinarily lie between the limits of 6 and 14 feet per second—depending on the height of the chimney and the temperature of the escaping gas—while for brick stacks the limits would be about 3 and 7 feet per second. Our previous investigations show, however, that such low velocities would usually be realized in brick stacks only when working below their economic capacity.

TEMPERATURE OF MAXIMUM DRAUGHT.

One of the problems often discussed in connection with the theory of chimneys, is that of finding the temperature of the escaping gases at which the weight of gas discharged per second will be a maximum. In order to ascertain this, we may combine Equations 2 and 4, which gives us

$$P = H\left(\frac{39}{T_a} - \frac{40}{T_s} - \frac{CMW^2T_s}{1,600A^3}\right). \quad . \quad . \quad . \quad (36)$$

From Equation 9, we have $BF = \frac{3,600\,W}{1.05}$, and substituting this value for BF in Equation 7, gives us

$$P = K\left(\frac{T_a W}{41a}\right)^2. \quad . \quad . \quad . \quad . \quad . \quad (37)$$

If we now place these two values of P equal to each other, and

then make the differential coefficient $\frac{dW}{dT_s}$ equal to zero, we shall obtain an equation to determine the temperature which will make W, or the weight of gas discharged per second, a maximum. Performing these operations, we have

$$T'_s = \frac{40}{39} T_a + \frac{40}{39} T_a \sqrt{1 + 0.9 \frac{KA^2 T_a}{CMHa^2}} \quad . \quad . \quad (38)$$

It must be remembered that the temperatures T_s and T_a are to be reckoned from the absolute zero, 461 degrees below the ordinary Fahrenheit zero.

It is evident from the last equation that the temperature of maximum draught is not invariable, but depends upon the dimensions of the chimney and furnace, different chimneys working best at different temperatures. It is also evident that this temperature can never be as low as that given by Rankine's formula before referred to, as that would require the second term under the radical sign to be equal to zero, which is not possible. The ordinary values of the temperature of maximum draught would range from 1,000 to 2,000 degrees on the common Fahrenheit scale—temperatures higher than it is desirable to attain in chimneys.

FORCED DRAUGHT.

When air is forced under the grate by means of a fan or steam jet, the resistance of the bed of coals does not need to be overcome by the chimney draught. It is usually desirable in such a case to have a chimney of sufficient draught to overcome the resistance offered by the combustion chamber, boiler-tubes, and flues; otherwise the pressure in the combustion chamber will be greater than that of the atmosphere, and the hot gases will tend to leak out of the furnace, or will rush out whenever the fire-door is opened. To design a chimney for use with forced draught, therefore, it is only necessary, in computing the value of K, or the coefficient of resistance for the furnace, to omit the first two terms of the second member of Equation 6, which terms correspond to the pressure required to produce the entrance velocity, and to overcome the resistance of the fire-bed.

This omission of terms, in the case of the furnace of which dimensions have been given, would reduce the resistance to be overcome by the chimney draught to about six-tenths of its original

value, and (by Equation 22) the height of the chimney needed with forced draught would be in this case six-tenths of that required for natural draught alone. On the other hand, by using a chimney of the same height as for natural draught, the use of the blast would allow the temperature of the escaping gases to be reduced to six-tenths of its original value on the common scale, thus effecting perhaps a saving of fuel. In some cases the diminution in the value of K would be greater than in the example cited, in other cases, less.

PROPORTIONS OF FURNACE, FLUES, ETC.

In proportioning furnaces, it is well to keep in view the object of making the coefficient of resistance, K, as small as practicable. For this reason, the access of air under the grate should be as direct and free as possible, the tube calorimeter should not be excessively contracted, $\left(\frac{a}{3}\text{ is a good value,}\right)$ and the passages leading from the boiler to the chimney should be as nearly circular as possible, smooth, with no sudden changes of cross section, sharp bends, or currents entering at right angles. The same rules used to determine the cross section of the chimney may be applied also to the horizontal smoke flues. A damper for regulating the draught should be placed either in the chimney or main flue, according to convenience, and, to obtain the best results, should be controlled by an automatic regulator.

MATERIAL AND SHAPE OF CHIMNEYS.

The principles developed in the foregoing discussion indicate that in the matter of draught there is very little choice between an iron and a brick chimney, the more rapid radiation of heat from the former being in a considerable degree offset by the greater frictional resistance and liability to leakage of the latter. The question as to which material to use should therefore be settled generally by considerations of cost, durability, and architectural fitness.

In regard to the shape of cross section of a chimney it may be said that the circular form is theoretically the best, on account of economy of material, and as offering the least resistance to the passage of the gases, and the least cooling surface. The difference between the draught obtained from a round chimney, and that

from a square one of the same height and sectional area, is, however, only one or two per cent., and brick chimneys may frequently be built more cheaply in the square form.

The best shape for a chimney cap is probably one that will deflect the wind upward, thus making it tend to help, instead of hindering, the draught.

The statement is sometimes met with that a better draught can be obtained if the chimney is not made uniform in its inside cross section, but tapering; some contending that the largest diameter should be at the bottom, others, that the top should be largest. In view of the fact that the total frictional resistance of a well-proportioned chimney is usually less than one-tenth of the resistance of the furnace, and in view of the very slight variation that can be made in the draught by a moderate change in the cross section of the chimney throughout its whole height, it is apparent that any improvement that might be made in the draught by tapering the chimney would be almost imperceptible; and while there might be no serious objection to the tapering form, its advantages, if any, would certainly not warrant any additional expense in construction.

In conclusion, it may be stated that the formulæ in this paper have been tested by comparison with the results obtained in a number of actual cases, and in so far as the writer has been able to observe, have been fairly sustained by the facts.

DISCUSSION.

*Mr. Geo. H. Babcock.**—This paper shows commendable zeal in investigating a problem which cannot be too well understood, and it develops some new ideas. Perhaps the most novel thing in it is the stated relation between the area and height of a stack for the most economical construction. If reliable, it is worth knowing. The method of arriving at the measure of friction in the stack is novel, and possibly nearer the truth than some of the former approximations. Whether the result is worth the amount of figuring necessary to apply it in practice may be questioned, however, when we remember how small a part of the whole it constitutes. Wiesbach gives it as 5%. The formulas given by Mr. Kent, in a paper† read before this Society, are simple and convenient for use, but do not take into account

* Contributed after adjournment.

† Transactions, Vol. VI., p. 81.

the friction of flues and furnaces, as in the formulas now given by Prof. Gale. But we are met at the outset with the difficulty that all such measures must necessarily be more or less indefinite. Every fact, however, which we can have recorded is valuable, and we have some in this paper worth noting.

Our author assumes that the draught of a chimney, as given by formula, has a fixed ratio to the number of pounds of fuel which can be burned therewith, without taking into consideration the many other controlling elements, as area of grate, kind and character of fuel, etc. Now, the fact is that the formulas, as usually given for chimneys, have little relation to that point, and cannot have, because there is such a wide difference between fuels and the conditions under which they are burned. A chimney which would burn a given quantity of "buckwheat" would burn more "lump" anthracite, still more bituminous coal, a still greater quantity of wood, and a vastly greater quantity of petroleum. Again, a chimney which would burn wood or coke to good advantage would not burn fine anthracite at all, and one which would burn bituminous coal freely would not suffice for culm. Hence this question is necessarily left out of consideration when treating of chimneys as chimneys.

But Prof. Gale has fallen into the same trouble with his intricate formula which he found with the old. He says (p. 2): "Some writers state this formula as if it were a matter of indifference in regard to draught; whether (other things remaining the same) we build a chimney 81 feet high and 2 feet square inside, or 16 feet high and 3 feet square inside, the product, $A \sqrt{H}$, being the same in each case. The 16-foot chimney would be considerably cheaper, and the main reason why it is not preferred is that practical engineers know that it is not an equivalent for the 81-foot chimney." Then, after he has evolved an intricate equation (12) to avoid this very objection, he says (p. 11): "It is evident from an inspection of this equation that, assuming different values for A, the sectional area, we may obtain an infinite number of chimneys of different heights, any one of which *will be capable of doing the work required.*" He also objects (p. 2) to the assumption that the area of chimney should be $\frac{1}{8}$ the grate area, but gives instead a rule (p. 13) that "the chimney area in square feet should equal the number of pounds of fuel to be burned per minute." This amounts to the same thing when the rate of combustion is

7.5 lbs. per hour per square foot of grate, and as the rate of combustion under the same chimney varies greatly with the kind of fuel, I fail to see wherein the last formula is any improvement.

But the principal contention in the paper is that the ordinarily received formulas for chimney efficiency are wrong. They are based on the Torricellian theory that $v = \sqrt{2gh}$, which has been proved correct for falling bodies and the flow of liquids. In chimneys, however, we are dealing with elastic fluids, and it has been established that there is a limit beyond which they apparently do not follow the same law as incompressible liquids; and if it can be shown that the flow in chimneys comes outside that limit, then it may be assumed that the formulas are wrong. But if the ordinary formulas are wrong in this respect, those given by Prof. Gale in this paper must be equally so, because they have as their basis the same Torricellian formula. But whether right or wrong is a matter of no consequence, as in boiler practice the extremes of temperature are limited by considerations of economy within moderate limits, and no engineer would think of carrying more than 600° up his stack for the purpose of producing draught, even if that were not the limit of chimney efficiency.

Is it a fact, however, that Prof. Gale is right in assuming that the formulas for the maximum discharge of air from a chimney are wrong? These formulas assume that the maximum flow of air in any chimney is, as stated by Rankine, " when the density of the hot gas is one-half that of the external air," which, of course, is to be understood as independent of friction either in chimneys, flues, or furnaces. If, however, we consider a chimney as a receiver having a reduced pressure into which the air is flowing from the atmospheric pressure without, according to the well-known law of elastic fluids first discovered by Napier, the maximum inflow will occur when the internal *pressure* is one-half that of the external; and, if there was no reduction of *density* from the increased temperature, this would agree with the rule as stated by Rankine, when the chimney was of the same height as the atmosphere. But there is a rapid decrease of density by the action of the same heat which produces the decrease of pressure, requiring the velocity of exit to be greatly increased over that of inlet—so much so that a maximum outflow is reached long before the inflow is at a maximum. For this *outflow* Ran-

kine's formula is correct, when the inflow is untrammelled, as he supposed it to be. It will be seen that, as the maximum outflow is at half the density of the external air, the area of inflow may be reduced to one-half the area of chimney, when the velocity will be the same as at the exit. If reduced below this, the velocity of influx will need to be increased by further increasing the head, in order to attain the maximum; and practically this is just what Prof. Gale supposes to be done by making the interstices between the coals on the grate so few and small that the effective area is less than half that of the chimney; and, of course, the temperature of the maximum flow is thus raised above Rankine's formula. This, however, does not prove the formula wrong, but rather sustains it. A practical point which may be gained by this discussion is, that in order that the maximum efficiency of a chimney may be attained, the area of openings through the fuel should practically be equal to half the chimney area, multiplied by the increase of volume of the air while passing through the fuel.

But this whole question of efficiency of chimneys is not amenable to exact analysis. There are elements always present which cannot be equated. For instance, the draught will always be affected more or less by the wind; generally—and particularly if the cap is of a proper form—it will be increased, but no man can tell how much. Again, when the atmosphere is quite still the heated air rising from the chimney will add an unknown and unknowable amount to its effective height. That a heated column without an enclosing chimney is effective is shown by every fire kindled in the open air, and also most forcibly by our whirlwinds and cyclones. Then I am inclined to think that the effect of infiltration of air through the brick walls of chimneys and flues has an influence which we have not generally accredited to it. To this is doubtless due the fact that iron stacks of the same area are more efficient than brick chimneys. I attach little importance to the effect of radiation, as the gases are usually within the chimney only from one to five seconds, and they cannot lose much heat in that time by radiation from the surface of the chimney.

Prof. C. H. Peabody.—In the discussion of this paper reference has been made to a theory of chimney draught given by Rankine, Péclet and others, as though it were the only theory that has been proposed. This theory, which gives a maximum draught at about 600° Fahr., was first proposed by Péclet in his

earlier works, from which it was taken by Rankine; but in his later works Péclet gives another theory, which does not give a maximum. There does not appear to be any practical advantage for one theory over the other.

Prof. J. Burkitt Webb.—My attention has been called to a criticism of Rankine and other noted authors, in a paper by Prof. Gale, in the discussion of which it was stated that one correct solution of the problem, if not two, is contained in *La Chaleur*, by Péclet. On examination it appears that the paper is based upon an incorrect hypothesis, and that the only solution in *La Chaleur* which is claimed therein to be correct gives the same result as does Rankine.

A brief comparison of the solutions will illustrate a point or two in mechanics, and show Rankine's deep mechanical insight. In Péclet's solution the pressure operating to produce the draught is found by subtracting the weight of a column of hot air, whose height is that of the chimney, from that of an equal column of cold air. The height of a column of air which will give this pressure is then calculated, and the velocity reckoned as due to this height—in the same manner as the velocity of water issuing from a vessel is deduced from its height therein. But Péclet hesitated between representing the pressure by a column of cold air and supposing it to produce the velocity with which the (cold) air enters the chimney, and representing it by a column of hot air producing the velocity of the (hot) air issuing from the top of the same.

M. Hudelo, editing the last edition of Péclet, says that the mechanical theory of heat furnishes us the means of deciding between the two formulas, and his analysis leads him to the choice of the column of hot air; the principles of thermo-dynamics, however, are not needed to show which formula should be used, the principles of mechanics known to Péclet being sufficient.

If the velocity be calculated by means of the column of cold air, the total pressure available to produce the draught will be appropriated to the production of the velocity of the entering (cold) air, and nothing will remain to cause the increase of velocity, which must accompany the increase of volume when the air is heated by the fire.

Evidently the pressure at disposal should be supposed to be occupied in causing the final velocity of the (hot) air issuing from the top of the chimney, unless the fire can be supposed respon-

sible for the increase of velocity as well as the increase of volume.

Now it is a principle in mechanics that the internal forces in a free body (or system of bodies) cannot affect the motion of the centre of gravity of the same ; and, therefore, when an elementary mass of air is heated, its internal expansive forces, acting equally in all directions, can have no effect upon the velocity of the mass, which is controlled entirely by external forces.

While it is true that the fire is responsible for the increase of velocity, and, indeed, for the whole velocity, it causes it in an indirect manner. In fact the fire simply keeps the clock wound up, so to speak, leaving it to run down under the control of gravity alone.

In refreshing contrast to the analysis in the last edition of Péclet, with its thermo-dynamic mistiness and approximations, is the clear and simple treatment in Rankine's *Steam Engine*, some pages before thermo-dynamics is reached. While Péclet is referred to, the analysis is given in Rankine's own way.

The question of head is disposed of in the following words :

"The formula (just given) enables the velocity U to be computed when the head N is given.

"The head N is expressed *in feet in height of a column of the hot gas in the chimney.*

"The head may be produced in three ways:

"I. By the draught of a chimney.

"II. By a blast-pipe.

"III. By a fan or other blowing machine.

"I. The head produced by the draught of a chimney is equivalent to the excess of the weight of a vertical column of cool air outside the chimney, and of the same height, above that of a vertical column of equal base, of the hot gas within the chimney; and, when expressed in *feet of hot gas*, it is found by computing the weight of a column of the cool external air as high as the top of the chimney is above the grate, and one foot square in the base, dividing by the weight of a cubic foot of the hot gas for the height of an equivalent column of hot gas, and subtracting the former height from the latter."

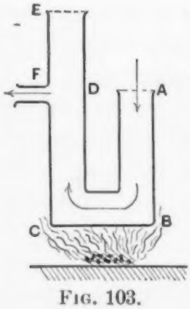


FIG. 103.

Fig. 103 illustrates Rankine's conception. CD is the chimney

whose draught is to be found. *DE* is a supposed extension, of such height that the column of hot gas *EC* exerts the same pressure per square foot as the column of cold air *AB*.

If the outlet at *F* is closed the column of hot gas balances statically that of cold air; but if *F* is open the hot gas flows out under the head *ED*, and if the air is supplied at *A* the flow will be from *A* to *F*.

As the atmospheric pressure is the same at *A* and *F* it can be omitted entirely from the problem.

Péclet starts with the supposition of a fire outside to warm the air passing from *B* to *C*, so that the chimney contains air alone; Rankine supposes hot gas in the chimney, but his analysis applies equally to hot air.

A model could be made to illustrate this action in a tube of the same form, *A B C D E*. Let water be supplied at *A*, and let a jet of air be blown in at right angles to the current between *B* and *C* with sufficient force to mix thoroughly with the water. In *AB* there will now be solid water and in *CE* a lighter mixture of water and air, which will rise to a height *DE* above the water-level at *A* when the mixture is properly proportioned, so that the issuing velocity at *F* will be that due to the head *DE*.

Thermo-dynamics may, however, be legitimately brought into the problem, for it must be evident that the density of the hot gas will not be the same at all parts of the chimney, and, to get at the weight of the column, some average density must be assumed or the law of its variation must be obtained upon thermo-dynamic principles. The former method is, however, sufficiently accurate for all purposes.

Rankine knew that thermo-dynamics was not needed in the problem, and he appreciated fully the importance in every problem of seeing clearly the exact bearing of the different principles of mechanics upon the subject, and of making use of such as lead to the simplest and most reliable conclusions.

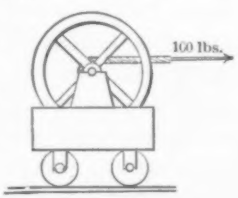


Fig. 104.

I am accustomed to use the following illustration of the independence of internal and external motions with my class in mechanics: Suppose a car (Fig. 104) to have mounted upon it a fly-wheel, on the shaft of which a rope is coiled. The rope is pulled in a direc-

tion parallel to the track with a constant force of 100 lbs. What difference will it make in the motion along the track whether the fly-wheel is free to turn, or whether it is blocked so that it cannot turn?

The principle of momentum solves the problem at once, for we have only to consider that the car and fly-wheel together constitute a body free to move horizontally, and that, so far as motion along the track is concerned, it is a matter of indifference whether there are also internal motions or not. The 100 lbs. will produce a linear momentum of 100 each second (neglecting friction) whether the fly-wheel is free to turn or not, so that the motion of the car will be the same as though the rope were fast to the car itself.

If, on the other hand, the problem be to find how much work the force does in each case, account must be taken of the fact that, when the fly-wheel is blocked, the 100 lbs. moves over the same space as the car, while, with the fly-wheel free, it will move over an additional space due to the rope unwinding from the axle. The simple principle of momentum used above is now no longer sufficient, and the angular momentum of the fly-wheel must be found, and the work done in rotating the wheel added to that done in moving the car and wheel along the track.

I propose, at the next meeting of the Society, to present a paper analyzing the thermo-dynamic solution in Péclet, which can hardly be considered as a rigorous proof of the formula, inasmuch as it departs considerably from the actual conditions of a chimney and does so in a way not likely to be noticed. This makes it necessary to show that such departure involves no essential error. What proof it does contain is, moreover, a repetition of previous analysis, and therefore superfluous.

Prof. J. E. Denton.—Referring to the foot of the eighth page of his paper, Mr. Gale states that no necessary relation is borne by the velocity in the chimney to that in the fire grate and tubes. This is true; but as soon as this relation is determined upon, as it must be for any given plant, there is a fixed relation, except for the element of temperature. This being so, every term in equation (6) will be expressible in terms of the velocity in the chimney, and the temperature of the latter times a series of constants. Hence the total head or pressure to produce the draught will be, as given by Rankine and Péclet,

$$P_r + P = \frac{V^2}{2g}(\alpha + \beta + \gamma, \text{etc.}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (\text{X.})$$

α, β, γ, etc., are constants representing the combined constants for temperature and velocity respectively. Any one term may be isolated, like P_f, as has been done by the author in equations (3) and (4). Placing these equations equal to each other, we have $W^2 = \alpha \frac{T_s^2}{V^2}$. Now the value of $P = Kv^2$, equation (7), should give rise to a precisely similar relation, but Mr. Gale has left out of this equation the value of the variable temperature, which is equal to T_s times a series of constants, as in (X.), the sum of which is represented in K. We should have, therefore, that the resistances represented by K are also controlled by a relation, $W^2 \alpha \frac{V^2}{T_s^2}$.

If, therefore, the temperature T_s was properly involved in equation (37) we should have the whole draught subject to the relation $W^2 \alpha \frac{V^2}{T_s^2}$, whence $W \alpha \frac{V}{T_s}$.

But $V = \sqrt{P + P_f}$, and by equation (2) $P + P_f = \alpha H\left(\frac{39}{40} \frac{T_s}{T_a} - 1\right)$.

Whence $W \alpha \frac{\sqrt{\frac{39}{40}(T_s - T_a)}}{T_s}$. If in this $\frac{39}{40}$ is called 0.96, the maximum value of W is identical with that in Rankine's *Steam-Engine* and in Péclet (edition 1878, Vol. I., art. 544), the result being that the draught is a maximum at about the temperature of melting lead.

Facts regarding chimneys, such as Mr. Gale makes the bases of his formulas (pages 2, 8 and 11), are greatly needed to render the subject available to formulas, and he should give them in full as he has observed them, if possible. It is to be regretted that the undoubted mathematical ability of the author has not had more extensive facts upon which to base constants. Péclet—from whose work all formulas for draught originate—had very meagre facts to deal with; but Péclet admits that practice can do better with empirical formulas than with the mathematical ones, until more facts or data are obtained. Mr. Gale, with less data than Péclet, asks practice to be guided by formulas considerably more intricate than those which Péclet pronounces rather impractical.

Mr. Kent's formula for chimneys, like many others, is quite satisfactory for practice in general; but it does not distinguish as to the true origin of the resistances, the greatest of which is the grate.

Mr. Gale's paper is valuable in emphasizing this point.

Prof. Peabody's remark makes me recall that there is such a solution as he describes in Péclet's treatise, in which the principles of thermo-dynamics are applied to the flow of gas in a chimney, as if the latter were a tube. The subject is so rough a one, however, from a mathematical standpoint, that Rankine's formulas are quite as practicable as any mathematical treatment need be.

On refreshing my acquaintance with Péclet, I find that the thermo-dynamic solution therein is the one which gives the same temperature for maximum draught as given by Rankine. (See Vol. I., pp. 539–40.)*

I hope Mr. Gale will especially add to his paper an account of the cases which he has observed which prove that chimney temperatures in excess of 600° cause increase of draught. Theory must always give way to facts derived from observation, and in a matter so very difficult of observation, as is the increase of draught due to changes of temperature and not to differences of obstruction, we should have the facts in full detail.

Prof. H. B. Gale.†—The author wishes in conclusion to present a few points in answer to the various criticisms which his effort has called forth. The objections may be best disposed of by considering them separately, in their order.

Mr. Babcock, after alluding to the author's "method of arriving at the measure of friction in the stack" as "novel, and possibly nearer the truth than some of the former approximations," questions "whether the result is worth the amount of figuring necessary to apply it in practice," etc. The ground of objection here seems hardly well chosen, inasmuch as no "figuring" at all is necessary to apply it in practice. A careful reading of the paper should have shown that in passing from equation 12 to equation 15 to find the most economical height for a chimney, the expression for friction in the stack reduces to a constant factor, whose value is included in the coefficient 100, in equation 22. In this respect, indeed, the latter formula has the advantage over those of Mr. Kent, which are recommended as "simple and convenient for use," in that his formulas *do* require a special calculation for the purpose of accounting for the element of chimney friction. The calculation is "simple" enough, to be sure, but as its sole basis is the arbitrary assumption that the retardation of velocity due to friction against the walls "is

* Added since the meeting. † Author's Closure, under the Rules.

equal to a layer of air two inches thick over the whole interior surface,"* it has a somewhat remote chance of being correct. In regard to the treatment of the matter of friction in the stack in this paper, the only "novel" feature consists in determining the coefficient of friction by direct measurements upon chimneys with the draught gauge, instead of relying upon values deduced from laboratory experiments on small tubes, as has been generally done. The expression for P_f in the paper may be derived from either of the expressions given respectively by Rankine, Péclet, and Weisbach, merely by making the requisite changes in notation. For example: in Article 233, page 287, of *The Steam Engine*, Rankine expresses the "head" required to overcome friction in the stack by $\frac{u^2}{2g}\left(\frac{fl}{m}\right)$ where u is "the velocity of the current in the chimney" (corresponding to V in the paper), f is a constant, l is the length of the chimney (here denoted by H), and m is "its area divided by its perimeter," or, in the notation of this paper, $\frac{A}{M}$. Making these changes in notation, Rankine's expression becomes $\frac{V^2}{2g}\frac{fHM}{A}$. On the same page in *The Steam Engine* we have Rankine's statement that the head "may be converted into an equivalent pressure in pounds on the square foot by multiplying by the density of the chimney gas." Therefore, representing that density by $\frac{40}{T_s}$, we obtain $P_f = \frac{40f}{2g}\frac{MHV^2}{T_s A}$, and massing the constants into one term, C, we have equation 3 of the paper. The value of the coefficient of friction in the stack, as directly measured by the methods described in this paper, is about 50% greater than the value "estimated" by Péclet for "sooty surfaces," and quoted by Rankine in the article referred to; a closer agreement than would be expected under the circumstances.

The next criticism charges me with assuming that "the draught of a chimney as given by formula has a fixed ratio to the number of pounds of fuel which can be burned therewith, without taking into consideration the many other controlling elements, as area of grate, kind and character of fuel, etc."

A glance at the paper will show clearly what assumptions are

* *Steam*, 20th edition, p. 60.

really made on this point, and that the "controlling elements" mentioned have not been neglected. Under *Design of Chimney to do Given Work*, we find "the number of pounds of coal which can be burned per second on a given grate is equal to the weight of air forced through [*i. e.*, the draught] divided by the number of pounds of air required for the combustion of a pound of coal *under the given conditions*." It is recognized that different furnaces and grades of fuel require different ratios of weight of air supplied to weight of coal consumed, and this variable ratio is represented by a factor B, which is carried through to equation 21. According to the excellent treatise on *Steam* already referred to the value of B may vary between the limits 12 and 29, though the extreme values are rarely approached in good practice ; but it is not assumed in the paper to be "a fixed ratio." In equations 22 to 25 the value 21 is used as an average, but following that is the statement that "in cases where the number of pounds of air supplied per pound of fuel differs much from the ordinary steam-boiler practice, we should go back to equation 21, *which takes account of these factors*."

As to the other "controlling element" mentioned, *viz.*, the "area of grate," it—or rather the area of *grate opening*, which is the more important feature as regards the draught—has been not only "taken into consideration," but appears as a factor a in every formula given by the author for the height of chimney or the weight of fuel consumed.

Now as to whether or not I have "a wrong conception of the application of the ordinary formula." The only application made of it in the paper, on which the charge must be based, is in the second section, where the equation $F = CA\sqrt{H}$ is cited "as an illustration" of the formulas derived from the common theory. Here F is the number of pounds of fuel that can be burned per hour, H the height and A the area of the chimney, and C a constant. An equation of this form is given by Weisbach, by Robert Wilson in his work on *Boiler and Factory Chimneys*, by Prof. C. A. Smith in his *Boiler Practice*, in Nystrom's *Mechanics*, in Haswell's *Engineers' Pocket Book*, and in a more or less modified shape by numerous other authorities, and will be allowed to be a fair illustration.* Some writers—among them Mr.

* Du Bois' Translation Weisbach's *Mechanics*, Vol. II., p. 194 ; *Boiler Chimneys*, Wilson [London, 1877] ; Smith's *Boiler Practice*, p. 20 ; Nystrom's *Mechanics*, Ed. 1882, p. 423 ; Haswell's *Engineers' Pocket Book*, Ed. 1885, p. 748.

Babcock, who, in *Steam*, has adopted Mr. Kent's formula—give the "horse-power" of the chimney instead of the number of pounds of coal burned per hour; but the principle is the same, as the horse-power in these cases is rated on the basis of the combustion per hour of a fixed weight of fuel.* Mr. Babcock, following Mr. Kent, further modifies the equation by using, instead of the area A, what he is pleased to call the "effective area," obtained by subtracting the equivalent of "a layer of air two inches thick" as allowance for friction. He rates a horse-power at 5 lbs. of coal per hour, and makes the power of the chimney $= 3.33E\sqrt{h}$, where E is the "effective area" and h is the height. I submit that all these formulas *are* open to the serious objection of purporting to give either the horse-power of a chimney or "the number of pounds of fuel which can be burned therewith, without taking into consideration the many other controlling elements, as area of grate," etc. Indeed, in the article by my critic, which assumes to give rules for the design of chimneys, no mention whatever is made of the "area of grate," and the reader is left to infer that it has no bearing on the problem! It appears to the writer that a formula which goes out of the way in an attempt to take into account the small element of friction in the stack, and then "necessarily leaves out of consideration" what our critic has called the "controlling elements," had better not be given, as it is certain to mislead whenever applied to conditions differing from the average.

The next argument (see paragraph beginning: "But Prof. Gale has fallen into the same trouble with his intricate formula which he found with the old," etc.) may be answered by simply asking those who are interested to compare equation 12 of the paper, to which my critic refers, with the equation $F = CA\sqrt{H}$. I think they will be found as essentially unlike as my rule that "the chimney area in square feet should equal the number of pounds of fuel burned per minute" is unlike the old rule which makes it "$\frac{1}{8}$ of the grate area" for all rates of combustion. On the latter point Mr. Babcock observes that the new rule amounts to the same thing as the old "when the rate of combustion is 7.5 lbs. per hour per square foot of grate." This is true; and he might have said also, with equal truth, that when the rate of combustion is *thirty* pounds per square foot per hour, it amounts to quite a different thing. The formulas for chimney area here

* See Mr. Kent's paper, *Transactions*, Vol. VI., p. 82.

presented are based on the assumption that the quantity of gas which a chimney has to discharge is proportional—roughly, of course—to the quantity of coal burned, and not to the area of the grate on which it is burned; and that consequently a chimney which is designed for free-burning bituminous coal needs a larger cross section for the same grate area than one designed to burn anthracite slack.

The remarks upon the theoretical formula, $v = \sqrt{2gh}$, applicable to the flow of incompressible fluids through frictionless orifices, are entirely beside the question—first, for the evident reason that a fire grate, furnace and chimney do not constitute a frictionless orifice, or anything like it; and second, because the deductions in the paper are not based upon this formula, as the critic supposes, but instead are based on the *experimental* law that the frictional resistance to the flow of a gas is proportional, approximately, to its density and to the square of its velocity (see section on *Determination of Resistance of Furnace*). The only place in the paper where the formula $v = \sqrt{2gh}$ is applied is in the approximate calculation of the coefficient of resistance for the furnace, K (equation 6), where the expression $2g$ appears in two terms, which are, however, relatively so small that they may be omitted without seriously affecting the result.

The discussion of the temperature of maximum draught by the same member is prefaced by the statement that it is a matter of no consequence, "as in boiler practice the extremes of temperature are limited by considerations of economy within moderate limits," etc. He has, nevertheless, thought it a matter of sufficient consequence in boiler practice to devote some space to it in a treatise of his own, where, as the final result of the investigation, it is stated that in the work of combustion "practically nothing can be gained by carrying the temperature of the chimney more than 350° above the external air at 60°." As this statement is not based upon experience, being simply a logical deduction from the common theory of draught, it might have been safer to say "*theoretically* nothing can be gained," etc. In point of fact, temperatures above this limit are more common in boiler chimneys throughout the United States than lower temperatures; where coal is cheap 600° and over is not unusual, and while the practice is often indefensible on grounds of economy, *practically* the higher temperatures *do* give a noticeable increase in the rate of combustion, or, what amounts to the same thing,

give an equally high rate with a chimney of less height.* But even were it admitted that in "boiler practice" the chimney temperature should never exceed 350°, the inference that the discussion of higher temperatures is a matter of no consequence would still be a *non sequitur*, as chimneys are used for other purposes, for some of which temperatures as high as 1,000° are employed.

Passing to the question, "Is it a fact, however, that Prof. Gale is right in assuming that the formulas for the maximum discharge of air from a chimney are wrong?" Mr. Babcock informs us that the common formulas are "to be understood *as independent of friction*, either in chimneys, flues or furnaces;" and again, "Rankine's formula is correct *when the inflow is untrammelled*, as he supposed it to be." With this rather important proviso I am quite willing to admit it to be correct, although Rankine says nothing about any such proviso. Now, in view of the fact—admitted by Rankine himself—that from 92 to 96% of the whole draught power of a chimney is employed in overcoming frictional resistances, it may fairly be asked how a formula derived "independent of friction" can possibly be otherwise than "wrong"? And when it is considered that at least ¾ of this resistance is usually encountered in the fire grate and boiler flues, is it not evident that the inflow to a chimney can hardly be described as "untrammelled"? Mr. Babcock admits that in certain cases "the temperature of the maximum flow is raised above Rankine's formula," but adds that "this does not prove the formula wrong." If he prefers the word "inapplicable," it will serve the purpose equally well.

In excuse for neglecting in his chimney formulas "the friction of flues and furnaces," my critic pleads the "difficulty" that such measures are necessarily inexact. Because a quantity cannot be determined with precision is it best to ignore it entirely, especially when it constitutes the main part of the problem? To put the case in a nutshell: According to the formula, $v = \sqrt{2gh}$, the velocity imparted to a current of air at 400° Fahr. by a head of ½ of an inch of water would be about 60 feet per second. The actual velocity of the gas in a chimney at that temperature, with the draught gauge showing a head of ½ an

* See *American Machinist*, Dec. 26, 1889, for interesting examples by W. E. Crane.

inch, is usually between 6 and 15 feet per second. Now, the difference of from 75 to 90% is due mostly to *friction*, and ¾ of that friction is in the fire grate and flues. It would seem to the ordinary person that this is an element of sufficient consequence to merit some attention.

The next point—that the draught of a chimney is liable to be increased by the wind, etc.—while not disputed, has no bearing on the design of a chimney (except, as stated in the paper, in regard to the form of the cap), as the chimney should be designed to give sufficient draught when the wind is too light to affect it appreciably. At other times the damper will take care of the increase.

Lastly, as to the effect of infiltration of air through the walls of brick chimneys and the small effect of radiation from iron ones, the author is very glad to be sustained in his position by an engineer of the standing and experience of Mr. Babcock.

Prof. Webb's comparison of the two "solutions" offered by Péclet for the theoretical velocity of a column of air, and his graphic description of Rankine's conception, are very clear and ingenious; but the bent tube pictured in Fig. 103 hardly represents the conditions of a boiler furnace, and when the frictional resistance of the fire-grate and flues also is omitted, the resemblance of the assumed problem to the practical one becomes still more remote. Nothing could be more apt than the illustration of his own method furnished by his solution of the problem of car traction ("neglecting friction"). The question as to which of Péclet's theories is correct, is like an inquiry whether an engineer would come nearer to the real force required to pull a cable car if ("neglecting friction") he assumed the rope to be attached to the car body, or to the hub of a wheel mounted thereon. On this question, the safest ground seems to be that taken by Prof. Peabody, that "there does not appear to be any practical advantage for one theory over the other."

As Prof. Webb has not touched upon the main problem of chimney draught, further comment upon his discussion is unnecessary; but it may be hoped that in his future treatment of the subject by thermo-dynamics, he will not neglect the "controlling elements."

In the mathematical discussion of Prof. Denton he appears to have discovered the real point at issue between the common

theory and that advanced in the paper—to wit, the method of calculating the *resistance of the furnace*. Since the greater part of the work of the draught is in overcoming this resistance, its determination is the most important problem in the theory of chimneys; and this member, as the only one of the critics to touch upon the subject, is entitled to the author's gratitude, although, in trying to elucidate the point, he falls into the same trap in which most writers have been caught, and succeeds only in emphasizing their blunder. Referring to the author's equation for the resistance of the furnace, $P = Kv^2$, in which v is the velocity of the entering air and K a coefficient for the furnace under consideration, the speaker says: "But Mr. Gale has left out of this equation the value of the variable temperature, which is equal to T_s times a series of constants."

In reply to this it need only be pointed out, as has been done already in the paper, that the frictional resistance encountered by the air in passing through the fire grate depends upon the velocity and temperature of the air *at that point;* and further, that neither the temperature of the entering air nor that of the fire in the furnace can be found by multiplying the variable chimney temperature T_s by a *constant*. The temperature of the air passing through the fire grate and bed of coals does not follow the variation of the chimney temperature, but is practically independent of it; and the same is true of the temperature at the entrance of the boiler tubes, the point next in importance to the fire grate as offering resistance to the passage of the gas. Hence, in investigating the effect of varying the chimney temperature T_s in the case of a given furnace, the temperatures at the points mentioned may be taken as approximately constant, and included in the coefficient K. In the expression for the pressure required to overcome the resistance of the furnace, therefore, the temperature T_s should not properly be involved, and our critic's defence of Rankine on this basis falls to the ground.

The absurdity involved in Rankine's method may be made more evident by an example. Let us suppose two chimneys of different diameters—one being, say, twice as great as the other—connected respectively to two furnaces of exactly the same dimensions, having grates of equal area, each burning, say, 22 lbs. of coal per square foot per hour, and discharging in the same time equal quantities of gas at the same chimney

temperatures, the height of the two chimneys being so adjusted as to give equal draught. It requires no very "deep mechanical insight" to perceive that equal furnaces, traversed by equal quantities of gas at the same temperature, will offer equal resistances, and that the pressures required to overcome those resistances must also be equal. This is the result given by the formula $P = Kv^2$, as the velocity of the entering air v will be the same in both cases, and the coefficients of resistance for the equal furnaces will also be equal. Now let us compare the furnace resistances in these cases, as calculated by Rankine's formula, given on page 287 of *The Steam Engine*, which represents the "head" required to overcome the furnace resistance by the expression $\frac{u^2}{2g} \times G$. In this formula G is stated to be "a factor of resistance for the passage of the air through the grate and the fuel above it, whose value, according to the experiments of Péclet on furnaces burning from 20 to 24 lbs. of coal per square foot of grate, is 12;" u is "the velocity of the current *in the chimney*." Now, as the cross section of one of our assumed chimneys is $\frac{1}{4}$ that of the other, and each discharges the same quantity of gas, the velocity u in the case of the former chimney will be 4 times as great as in the case of the larger one; and the other quantities in the expression being constant, we shall have the "heads" required to balance the friction of the two furnaces respectively proportional to the values of u^2 in the two cases. Moreover, as the head may be converted into an equivalent pressure by multiplying by the density of the chimney gas, and the density of the gas in the two chimneys is the same, we obtain the result that the pressure required to balance the resistance of one of our given furnaces is just *sixteen times as great* as that required to balance the resistance offered by an equal furnace to the passage of the same quantity of gas at the same temperature!

It devolves upon those who maintain that "Rankine's formulas are quite as practicable as any mathematical treatment need be" to explain this apparent discrepancy.

It is stated in the discussion that "Mr. Gale, with less data than Péclet, asks practice to be guided by formulas considerably more intricate than those which Péclet pronounces rather impractical." In regard to the first point, the author would say that all Péclet's data were freely used in writing the paper for

purposes of comparison, in addition to the material derived from original experiments; and in regard to the second, he invites comparison of formulas 20 and 22—which are those chiefly recommended for practice—with the formulas of Péclet and Rankine. Such a comparison will show that the author's formulas are no more intricate, and—he is confident—are somewhat less "impractical."

CCCLXXVII.

GENERAL SOLUTION OF THE TRANSMISSION OF FORCE IN A STEAM ENGINE, AS INFLUENCED BY THE ACTION OF FRICTION, ACCELERATION, AND GRAVITY.

BY D. S. JACOBUS, HOBOKEN, N. J.
(Member of the Society.)

As there is a general tendency among mathematicians to avoid all equations that contain approximations, this subject has been worked over many times, each writer trying to develop equations more general than those previously obtained, notwithstanding the fact that many of these same equations solve the problem with all the exactness that will ever be required for practical work.

Solutions of this problem have been given by other members of this society, and others in this country, some of which have been admirable; the subject has also been taken up in a very elaborate approximate way by many German writers. These solutions have been thoroughly looked into during the last two years, in which the writer has worked at intervals on the problem.

It is endeavored in this paper to present a set of equations involving every condition met with in ordinary engine practice, not as a set for every-day use, but valuable in showing that the more approximate ones are sufficiently accurate for ordinary purposes. These equations are also so general that it is hoped that persons of mathematical tendencies will no longer feel that there is any important error involved, and try to develop a set of a more exact nature.

Most writers on this subject have been satisfied with simply determining the accelerating forces, and not showing how these forces modify the tangential effort transmitted to the crank, the pressures on the crank and wrist-pins, and the shaking forces acting on the bed; in the present article, however, this modification will be discussed and numerical examples given.

The accelerating forces must necessarily be used in determining the following quantities:

1st. The pressures that exist at the bearings.

2d. The effort transmitted to the crank at each point of its revolution, knowing which the fluctuation of energy can be determined.

3d. The forces tending to shake the bed of the engine.

The general and approximate equations have been applied to the following four classes of engines, and the difference in the results determined.

(*a*) Small horizontal high-speed engine.

(*b*) Locomotive.

(*c*) Slow speed of revolution. Harris—Corliss.

(*d*) Engine in which the line of travel of the wrist-pin does not pass through the centre of the crank shaft. Westinghouse.

The main results of the calculations are given in Table I.

The quantity $\frac{\Delta E}{\int P ds}$, as given in Table I., is that employed by Rankine in determining the proportions of fly-wheels proper for given engines; it is the ratio of the periodical excess or deficiency of energy ΔE, to the whole energy exerted per revolution, $\int P ds$. The curves of effort transmitted to the crank shaft, together with diagrams representing the magnitude and lines of action of the pressures acting on the crank-pin, are given in Figs. 52 to 65.

The following conclusions may be drawn from the results contained in Table I.:

1st. The greatest difference of the results by the exact and approximate methods for the maximum crank-pin pressures is 2.3 per cent.

2d. The greatest difference between the values of $\frac{\Delta E}{\int P ds}$ by the two methods is 1.3 per cent.

Tables VI. to X. contain the numerical results that are employed in laying off the curves in the figures previously mentioned. In these tables the forces are given in pounds per square inch of piston area, so that they may be readily laid off on the indicator diagrams. The curves for the Westinghouse engine are determined graphically, and therefore no table is given for this case.

In making use of the shaking forces in order to determine the proper size of counterweight, we have a very complex problem to deal with. In this article it is not attempted to show anything farther than the magnitude and direction of these forces in a given

TABLE I.

RESULTS OF NUMERICAL APPLICATIONS TO SEVERAL STANDARD CLASSES OF ENGINES.

Class of Engine.	Conditions assumed.	Dimensions. Bore.	Dimensions. Stroke.	Revolutions per minute.	Indicated horse power.	$\frac{\Delta E}{\int P ds}$ Exact.	$\frac{\Delta E}{\int P ds}$ Approximate.	$\frac{\Delta E}{\int P ds}$ Not including accelerating forces.	Pressure acting on crank-pin. Exact.	Pressure acting on crank-pin. Approximate.	Pressure acting on crank-pin. Not including accelerating forces.	Number of table containing results of calculations.
		Inches.	Inches.									
Small horizontal high-speed........	Not including the effects of friction and weight..........................	10	12	300	57	.184	.174	.238	58.1	57.3	83.0	VI.
Small horizontal high-speed........	Not including friction but including weight...	10	12	300	57	.180	....	....	58.0	... .	....	VII.
Small horizontal high-speed.	Including both friction and weight.....	10	12	300	57	.171	....	....	56.8	....	....	VIII.
Locomotive........................	Not including friction and weight.	$18\frac{1}{4}$	24	250	*345	.181	.172	.275	56.8	55.5	108.0	IX.
Corliss. Slow speed of revolution..	Not including friction and weight......	$26\frac{1}{4}$	60	60	346	.142	.131	.185	76.1	75.5	89.4	X.
Westinghouse.....................	Not including friction and weight......	11	10	320	66	.160	.147	.247	61.0	60.8	75.5	......

* For one cylinder.

engine for different sizes of counterweight. The shaking effect of these forces is not treated in a general way, which, if done, involves numerous complications, such as the expression of the force with which the foundation will resist springing in both a horizontal and vertical direction, and the determination of whether the time of revolution of the engine will harmonize with the vibrations of its bed. The simplified problem of determining the movement of an engine supposed to be suspended in space, has been worked out by the writer, and will shortly be published; but this, of course, is not the condition of an engine set in the ordinary way, and it cannot be predicted that the engine having the least shake under these assumed conditions will behave the best when secured to a firm foundation.

The translative shaking forces acting on the bed of a horizontal high-speed engine, for different counterweights, are shown in magnitude and direction in Fig. 66, the curves there given being drawn through the ends of the lines representing the forces.

It will be seen in Fig. 66 that the curves drawn through the extremities of the shaking forces are not ellipses, as is indicated in some of the more approximate solutions of this subject that have been offered by other writers.

The conclusions that may be drawn in regard to the approximate and exact formulæ are,

1st. The approximate formula gives results that are exact enough to be used in determining the ratio $\frac{\Delta E}{Pds}$ for fly-wheel work, for, although this quantity differs somewhat by the two methods, an assumption of a change in the steam pressure or of the total work done by the engine will cause a greater variation in ΔE than there is between the approximate and exact results; and as the fly-wheel must be made heavy enough to take care of these changes, it is not necessary to determine ΔE so closely that the wheel may be exactly proportioned to a particular power and shape of indicator card when a change in these will alter the final result.

2d. The approximate and exact methods are seen to agree very closely in regard to the maximum pressures on the journals, so that for all practical work one is as good as the other in determining this quantity.

3d. In determining the size and position of the counterweight we have to make use of the exact formulæ in order to determine

the forces tending to shake the bed. In the case of the horizontal high-speed engine, for which the shaking forces are shown in Fig. 66, it may be seen that the sum of the vertical shaking forces is very nearly balanced by a counterweight of one quarter the mass of the reciprocating parts placed opposite the crank-pin at the same radius as the latter; and those in a horizontal direction are best balanced by the entire mass at the same point. It therefore appears that for this particular engine the smaller counterweight will be the best if the shake occurs more readily in a vertical than in a horizontal direction, and that the heavier one will be preferable when the horizontal forces produce the most shake, as, for instance, if the engine is set on a tall foundation.

In * the Appendix are given the equations employed in calculating these results, together with abstracts of the methods of deriving them.

It may be well to note that the accelerating forces given for the rod are such as will produce the required acceleration of the same. While there is but one *single* force, with its special point of application, that will produce the required acceleration, there are an infinite number of sets of forces at the wrist and crank-pins that effect the same results, these sets differing only in the amount of tension or compression which they produce in the rod. This is the cause of the apparent disagreement of many of the formulæ presented by different writers. All correct sets will, however, give the true values of forces acting between different parts of the engine, such as the crank and wrist-pin pressures, tangential effort, etc. This class of formulæ cannot be employed to find the internal strains without a complete discussion of the forces acting on each element of the rod.

APPLICATION OF THE FORMULÆ.

Table IV., in connection with the indicator diagrams shown in

* The method of obtaining the accelerating forces has been indicated in an abstract presented to the American Association for the Advancement of Science, and will be published in detail in the *Annals of Mathematics*, which paper has already contained an article by Professor J. Burkitt Webb and the writer, on the effect of friction at connecting-rod bearings on the forces transmitted.

The writer is much indebted to Professor Webb for valuable assistance in the present article, many of the equations having been worked out independently by him and compared, in order to check the final results. The methods employed were also, in a great measure, suggested by him, as well as a portion of the notation.

Figs. (52), (56), (59) and (62), gives the data which are required to calculate the accelerating forces, rotative effort, etc., for several typical classes of engines. Making use of the data given in Table IV., we obtain the constants for the equations of the accelerating forces given in Table V. These constants are used in calculating the results given in the tables that follow.

The effect of friction and the action of gravity is neglected in Tables VI., IX., and X. In Table VII. the action of gravity is included, but not that of friction. Table VIII. gives the effect of friction at the wrist and crank-pins, together with that of the action of gravity.

We have taken the co-efficient of friction at the wrist and crank-pins in Table VIII. equal to six per cent.*

TABLE II.

VALUES OF Z WHEN THE LINE OF TRAVEL OF THE WRIST-PIN PASSES THROUGH THE CENTRE OF THE CRANK SHAFT.

Crank angle = θ.	Ratio of connecting-rod to radius of crank circle = n.			
	4	5	6	$7\frac{2}{3}$
0, 180, 360.....	+ .250	+ .200	+ .167	+ .130
10, 170, 190, 350	+ .236	+ .188	+ .157	+ .123
20, 160, 200, 340.....	+ .194	+ .154	+ .128	+ .100
30, 150, 210, 330.....	+ .129	+ .102	+ .084	+ .066
40, 140, 220, 320.....	+ .048	+ .037	+ .030	+ .023
50, 130, 230, 310.....	− .040	− .033	− .028	− .022
60, 120, 240, 300.....	− .125	− .100	− .083	− .065
70, 110, 250, 290.....	− .195	− .155	− .129	− .100
80, 100, 260, 280.....	− .242	− .191	− .159	− .124
90, 270, ...	− .258	− .204	− .169	− .132

* In experiments made at the Stevens Institute a great variation of co-efficients of friction have been observed with different conditions of journals, the range being from one-half of one per cent. to eight per cent.

TABLE III.

VALUES OF Z WHEN $n = 5$, $b = \frac{1}{2}$, AND $a = 0$.

θ	Z	θ	Z
0	+.2030	180	+.2030
10	+.2066	190	+.1758
30	+.1500	210	+.0574
50	+.0422	230	−.1092
70	−.0593	250	−.2559
90	−.1005	270	−.3145
110	−.0593	290	−.2559
130	+.0422	310	−.1092
150	+.1500	330	+.0574
170	+.2066	350	+.1758

TABLE IV.

DIMENSIONS AND SPEED OF ENGINES TO WHICH THE FORMULÆ HAVE BEEN APPLIED.

	Class of Engine.				
	Small horizontal high-speed.	Large horizontal high-speed. Locomotive.	Large horizontal slow-speed of revolution. Harris-Corliss.	Westinghouse.	Notation in Formulæ.
Revolutions of crank shaft per minute	300	250	60	320	$30\frac{r}{\pi}$
Length of stroke, in inches	12	24	60	10	
Diameter of cylinder, in inches	10	$18\frac{1}{2}$	$26\frac{1}{8}$	11	
Length of connecting-rod, in inches	36	92	150	$5R$	nR
Distance from the wrist-pin to the centre of gravity of rod, in inches	20.15	55	66	$3.22R$	lR
Distance from the centre of the crank shaft to the line of travel of the wrist-pin, in inches	0	0	0	$0.5R$	bR
Principal radius of gyration, in inches	15.00	34.1	48	$2.07R$	kR
Weight of piston, piston-rod, and cross-head, in pounds	90.	474	1300	100	W
Weight of connecting-rod, in pounds	70.	307	1200	50	W_2^1

TABLE V.

VALUE OF THE CONSTANTS EMPLOYED IN THE EQUATIONS.

Class of Engine.	n	l	k	τ	F_1	Y_1	Y_2	X_1	X_2
Small horizontal high speed	6	3.36	2.50	31.42	$17.57 (\cos\theta + Z)$	$-.99 \sin\theta$	$-6.66 \sin\theta$	$6.01 \cos\theta + 5.02 Z$	$7.65 \cos\theta + .99 Z$
Large horizontal high speed. Locomotive.	$7\frac{2}{3}$	4.58	2.84	26.18	$37.54 (\cos\theta + Z)$	$-2.51 \sin\theta$	$-12.01 \sin\theta$	$9.79 \cos\theta + 7.28 Z$	$14.52 \cos\theta + 2.51 Z$
Large horizontal slow speed of revolution Harris—Corliss....	5	2.6	1.6	6.3	$7.42 (\cos\theta + Z)$	$-1.01 \sin\theta$	$-2.55 \sin\theta$	$3.29 \cos\theta + 2.28 Z$	$3.56 \cos\theta + 1.01 Z$
Westinghouse	5	3.22	2.07	33.51	$15.20 (\cos\theta + Z)$	$-.44 \sin\theta$	$-4.45 \sin\theta$	$2.71 \cos\theta + 2.27 Z$	$4.90 \cos\theta + .44 Z$

TABLE VI.

SMALL HORIZONTAL HIGH-SPEED ENGINE. NOT INCLUDING THE EFFECTS OF FRICTION AND GRAVITY. FORCES IN LBS. PER SQUARE INCH OF PISTON AREA.

	By Exact Formulæ.										By Approximate Formulæ.			Inertia not Considered.	
θ	F_1	Y_1	Y_2	X_1	X_2	P_p	P_a	T	P_c	P_w	P_p	T	P_c	T	P_c
0	+20.50	0	0	+6.85	+7.82		+83	0	+47.8	+62.5	+36.4	0	+46.6	0	+83.0
10	+20.06	−.17	−1.16	+6.71	+7.69	+27.77	+82	+11.0	+47.7	+61.9	+35.6	+ 9.4	+46.4	+16.6	+82.0
20	+18.76	−.34	−2.28	+6.29	+7.32	+25.97	+82	+22.2	+49.9	+63.3	+33.3	+19.3	+48.7	+32.4	+82.1
30	+16.69	−.50	−3.33	+5.63	+6.71	+23.14	+81	+33.1	+52.6	+64.4	+29.7	+29.4	+51.5	+46.4	+81.3
40	+13.99	−.64	−4.28	+4.75	+5.89	+19.44	+79	+43.2	+55.4	+65.3	+24.8	+39.3	+54.5	+57.3	+79.5
50	+10.81	−.76	−5.10	+3.72	+4.89	+15.08	+76	+51.7	+58.1	+65.6	+19.2	+48.2	+57.3	+64.5	+76.6
60	+ 7.33	−.86	−5.77	+2.59	+3.74	+10.30	+68	+54.2	+56.2	+61.0	+13.0	+51.6	+55.6	+63.8	+68.7
70	+ 3.74	−.93	−6.26	+1.41	+2.49	+ 5.35	+54	+48.4	+48.4	+50.7	+ 6.7	+47.0	+47.9	+53.7	+54.7
80	+ .26	−.97	−6.56	+ .25	+1.17	+ 3.52	+43	+43.1	+43.5	+43.2	+ .5	+43.1	+43.1	+43.6	+43.6
90	− 2.97	−.99	−6.66	− .85	− .17	− 8.99	+33	+37.0	+39.2	+36.4	− 5.3	+38.3	+38.8	+33.0	+33.5
100	− 5.85	−.97	−6.56	−1.84	−1.49	− .03	+28	+34.4	+39.2	+34.2	−10.4	+36.7	+38.9	+26.8	+28.4
110	− 8.28	−.93	−6.26	−2.70	−2.74	−11.47	+20	+27.9	+35.5	+28.6	−14.7	+30.7	+35.1	+17.7	+20.2
120	−10.24	−.86	−5.77	−3.42	−3.91	−14.29	+11	+20.1	+30.1	+21.5	−18.2	+23.2	+29.5	+ 8.7	+11.1
130	−11.79	−.76	−5.10	−4.00	−4.95	−16.54	+ 1	+12.0	+22.9	+12.9	−20.9	+15.0	+22.1	+ .7	+ 1.0
140	−12.93	−.64	−4.28	−4.45	−5.83	−18.22	−15	+ 1.8	+ 9.4	+ 2.1	−23.0	+ 4.5	+ 8.0	− 8.4	−15.1
150	−13.74	−.50	−3.33	−4.78	−6.54	−19.42	−25	− 2.4	− 2.7	−11.3	−24.4	− 3.0	− .6	−10.7	−25.1
160	−14.27	−.34	−2.28	−5.01	−7.06	−20.22	−36	− 4.6	− 9.8	−21.8	−25.3	− 3.1	−10.7	−10.4	−36.1
170	−14.55	−.17	−1.16	−5.13	−7.38	−20.64	−42	− 3.1	−14.9	−27.5	−25.9	− 2.3	−16.1	− 6.1	−42.0
180	−14.64	0	0	−5.17	−7.48		−80	0	−52.7	−65.4	−26.0	0	−54.0	0	−80.0

TABLE VII.

SMALL HORIZONTAL HIGH-SPEED ENGINE. ACTION OF GRAVITY INCLUDED, BUT NOT THAT OF FRICTION. FORCES IN LBS. PER SQUARE INCH OF PISTON AREA.

θ	G	T	N	P_c
0	+ .4	− 0.5	+48.3	+48.3
10	+1.8	+10.5	+46.5	+47.6
30	+4.8	+32 7	+41.2	+42.5
50	+7.6	+51.4	+26:8	+58.0
70	+7.2	+48.2	+ 3.2	+48.3
90	+5.6	+39.0	−12.4	+40.9
110	+4.4	+28.1	−21.6	+35.4
130	+1.8	+12.3	−19.1	+24.8
150	− .6	− 2.0	− 1.2	− 2.3
170	− .4	− 2.6	+14.7	−14.9
180	+ .4	+ .5	+52.9	−52.9

TABLE VIII.

SMALL HORIZONTAL HIGH-SPEED ENGINE. FRICTION AND ACTION OF GRAVITY INCLUDED. FORCES IN LBS. PER SQUARE INCH OF PISTON AREA.

θ	A	B	G_f	N_f	T'_f	P_{cf}	P_{wf}
0	+.13	+.12,	+ .1	+46.1	− 1.5	+46.4	+60.8
10	+.13	+.12!	+1.5	+44.8	+ 9.5	+45.9	+60.3
30	+.13	+.13	+4.4	+39.8	+30.7	+50.9	+62.7
50	+.14	+.15	+7 0	+26.1	+49.0	+56.8	+63.8
70	+.11	+.12	+6.8	+ 3.1	+46.2	+47.6	+49.4
90	0	+.10	+5.3	−12.2	+35.7	+38.4	+35.5
110	−.06	+.09	+4.1	−21.3	+27.2	+34.8	+28.1
130	−.03	+.06	+1.8	−19.0	+12.0	+22.5	+12.8
150	−.02	+.01	− .6	− 1.2	− 2.0	− 2.3	−10.8
170	−.06	+.04	− .4	+13.8	− 2.7	−14.1	−26.6
180	−.14	+.13	+ .4	+51.2	− 0.3	−50.9	−63.8

TABLE IX.

LOCOMOTIVE ENGINE. NOT INCLUDING THE EFFECTS OF FRICTION AND GRAVITY. FORCES IN LBS. PER SQUARE INCH OF PISTON AREA.

	By Exact Formulæ.									By Approximate Formulæ.			Neglecting Inertia.	
θ	F_1	Y_1	Y_2	X_1	X_2	P_p	P_a	T	P_c	F_p	T	P_c	T	P_c
0	+42.42	0	0	+10.74	+14.85		+108	0	+40.0	+69.9	0	+38.1	0	+108.0
10	+41.59	− .44	− 2.09	+10.54	+14.61	+54.6	+108	+10.5	+41.4	+68.5	+ 7.8	+39.5	+21.3	+108.0
20	+39.03	− .86	− 4.11	+ 9.93	+13.90	+51.5	+107	+21.5	+44.7	+64.3	+16.0	+42.7	+41.8	+107.1
30	+34.99	−1.26	− 6.01	+ 8.96	+12.74	+46.0	+102	+31.3	+46.4	+57.6	+24.8	+44.5	+57.1	+102.2
40	+29.62	−1.61	− 7.72	+ 7.67	+11.18	+39.1	+ 95	+39.7	+48.3	+48.8	+32.5	+46.3	+67.5	+ 95.3
50	+23.30	−1.92	− 9.22	+ 6.13	+ 9.28	+30.8	+ 86	+46.0	+49.7	+38.4	+39.6	+47.8	+73.1	+ 86.5
60	+16.33	−2.17	−10.40	+ 4.42	+ 7.10	+21.8	+ 80	+53.9	+52.4	+26.9	+49.2	+53.5	+74.8	+ 80.6
70	+ 9.09	−2.36	−11.29	+ 2.62	+ 4.72	+12.2	+ 70	+56.9	+56.8	+15.0	+54.2	+55.5	+68.9	+ 70.6
80	+ 1.86	−2.47	−11.84	+ .80	+ 2.21	+ 2.8	+ 56	+53.7	+54.6	+ 3.1	+53.3	+53.3	+56.3	+ 56.4
90	− 4.95	−2.51	−12.01	− 9.6	− .33	− 6.2	+ 45	+51.2	+54.7	− 8.2	+53.2	+53.7	+45.0	+ 45.4
100	−11.17	−2.47	−11.84	− 2.60	− 2.83	−14.5	+ 36	+48.6	+55.8	−18.4	+51.9	+54.9	+27.4	+ 28.8
110	−16.59	−2.36	−11.29	− 4.08	− 5.22	−21.7	+ 20	+37.4	+48.8	−27.3	+42.3	+47.6	+18.2	+ 20.2
120	−21.21	−2.17	−10.40	− 5.37	− 7.42	−28.1	+ 7	+28.3	+43.5	−34.9	+33.8	+42.2	+ 5.6	+ 7.0
130	−24.96	−1.92	− 9.22	− 6.45	− 9.39	−33.2	− 8	+17.6	+25.6	−41.0	+23.4	+33.4	− 5.7	− 8.1
140	−27.89	−1.61	− 7.72	− 7.33	−11.07	−37.3	− 27	+ 5.9	+21.1	−46.0	+11.0	+19.1	−15.8	− 27.1
150	−30.03	−1.26	− 6.01	− 8.00	−12.41	−40.3	− 43	− 1.2	+ 9.3	−49.5	+ 2.9	+ 6.5	−18.9	− 43.1
160	−31.52	− .86	− 4.11	− 8.47	−13.39	−42.7	− 69	− 7.8	−15.8	−52.0	− 5.1	−16.9	−20.9	− 68.9
170	−32.35	− .44	− 2.09	− 8.75	−13.99	−43.6	− 88	− 6.7	−32.9	−53.3	− 5.2	−34.7	−13.2	− 88.0
180	−32.66	0	0	− 8.84	−14.19		−103	0	−47.3	−53.8	0	−49.2	0	−103.0

TABLE X.

LARGE HORIZONTAL ENGINE SLOW-SPEED OF REVOLUTION. NOT INCLUDING EFFECTS OF FRICTION AND GRAVITY. FORCES IN LBS. PER SQUARE INCH OF PISTON AREA.

	By Exact Formulæ.									By Approximate Formulæ.			Neglecting Inertia.	
θ	F_1	Y_1	Y_2	X_1	X_2	P_p	P_a	T	P_c	P_p	T	P_c	T	P_c
0	+8.9	0	0	+3.7	+3.8		+86	0	+69.6	+17.2	0	+68.8	0	+86.0
10	+8.7	− .2	− .4	+3.7	+3.7	+13.4	+87	+15.3	+71.7	+16.8	+14.6	+70.3	+18.1	+87.1
30	+7.2	− .5	−1.3	+3.1	+3.2	+11.1	+89	+45.8	+76.1	+13.8	+44.1	+75.5	+52.2	+89.4
60	+3.0	− .9	−2.2	+1.4	+1.7	+ 4.8	+46	+39.3	+41.4	+ 5.7	+38.4	+40.9	+43.9	+46.7
90	−1.5	−1.0	−2.6	− .5	− .2	− 2.2	+21	+23.2	+24.3	− 2.9	+23.9	+24.4	+21.0	+21.4
120	−4.5	− .9	−2.2	−1.9	−1.9	− 7.0	+13	+15.6	+21.9	− 8.6	+16.8	+21.9	+10.1	+13.2
150	−5.7	− .5	−1.3	−2.6	−3.0	− 9.2	+10	+ 7.9	+21.5	−10.9	+ 8.6	+21.0	+ 4.3	+10.0
170	−5.9	− .2	− .4	−2.8	−3.3	− 9.7	+ 5	+ 2.0	+17.0	−11.4	+ 2.3	+16.4	+ .7	+ 5.0
180	−5.9	0	0	−2.8	−3.4		−85	0	+72.9	−11.4	0	−73.6	0	−85.0

TABLE XI.

SHAKING FORCES ACTING ON THE BED OF A SMALL HIGH-SPEED HORIZONTAL ENGINE WITH NO COUNTERWEIGHT. FOR DIMENSIONS, SEE TABLE IV. FORCES IN LBS. PER SQUARE INCH OF PISTON AREA.

θ	X_t	Y_t	θ	X_t	Y_t	θ	X_t	Y_t
0	− 36.0	0	120	+ 17.9	+ 6.63	240	+ 17.9	− 6.63
10	− 35.2	+ 1.32	140	+ 23.0	+ 4.91	260	+ 9.9	− 7.53
20	− 33.0	+ 2.61	160	+ 25.7	+ 2.61	270	+ 4.8	− 7.65
40	− 24.8	+ 4.91	170	+ 26.3	+ 1.32	280	− .9	− 7.53
60	− 13.2	+ 6.63	180	+ 26.5	0	300	− 13.2	− 6.63
80	− .9	+ 7.53	190	+ 26.3	− 1.32	320	− 24.8	− 4.91
90	+ 4.8	+ 7.65	200	+ 25.7	− 2.61	340	− 33.0	− 2.61
100	+ 9.9	+ 7.53	220	+ 23.0	− 4.91	350	− 35.2	− 1.32

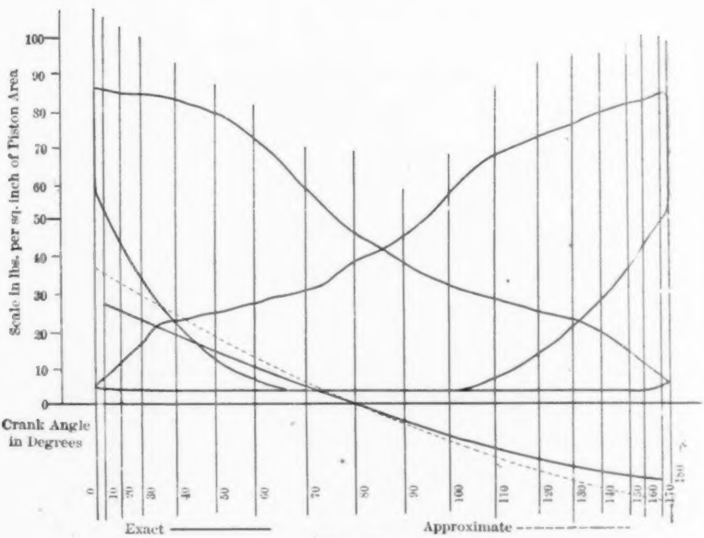


FIG. 52.
Small Horizontal High Speed Engine.

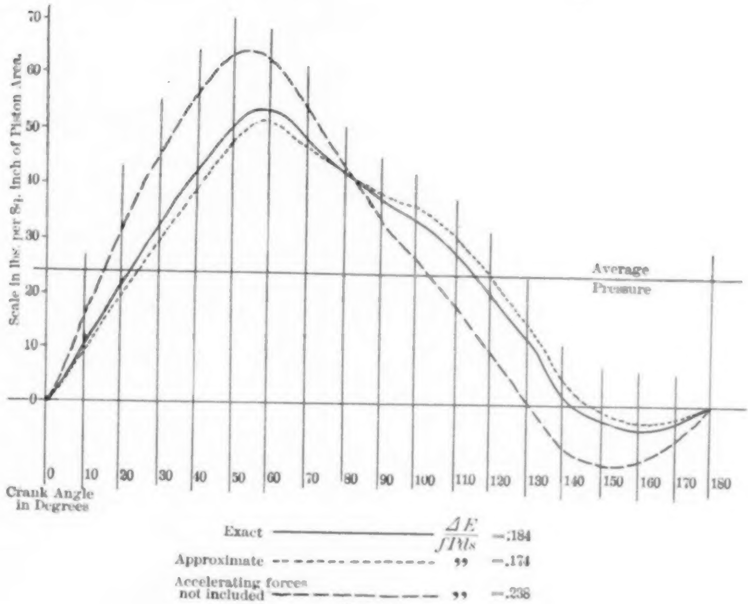


FIG. 53.

Small Horizontal High Speed Engine.—Tangential Effort Acting on Crank-Pin, Friction and Gravity Not Included.

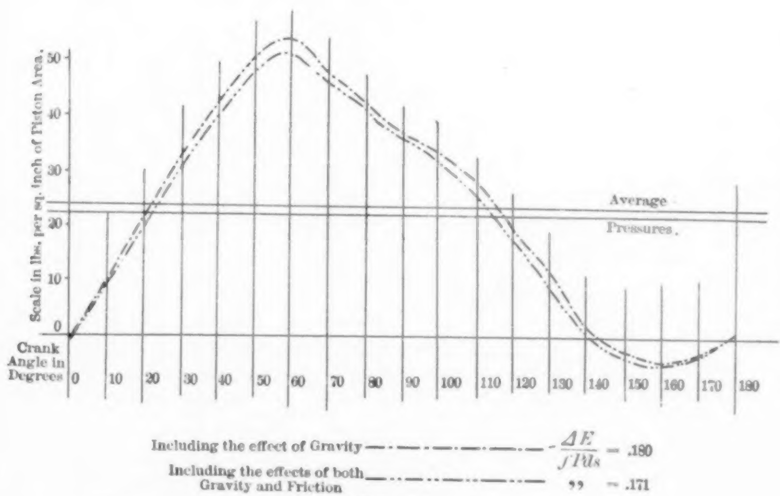


FIG. 54.

Small Horizontal High Speed Engine.—Tangential Effort Acting on Crank-Pin. Accelerating Forces Determined by Exact Method, and Friction and Gravity Included.

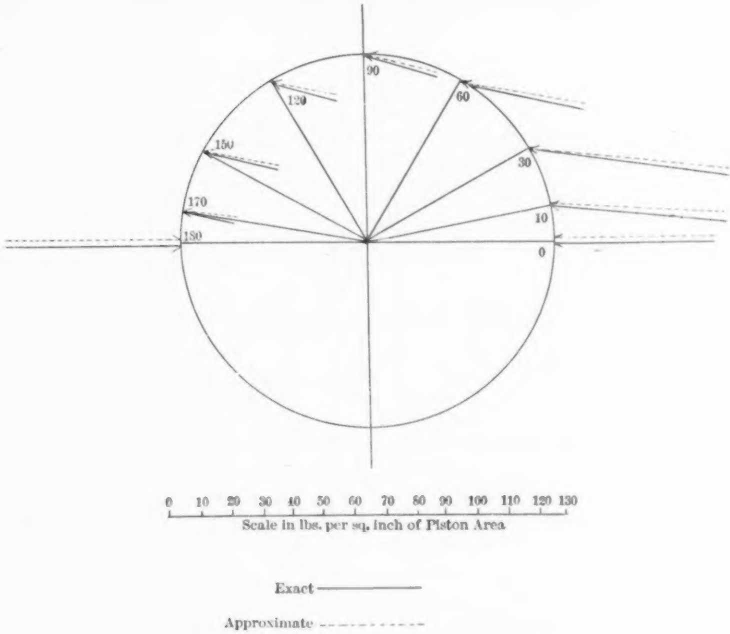


FIG. 55.
Small Horizontal High Speed Engine.—Pressures Acting on Crank-Pin.

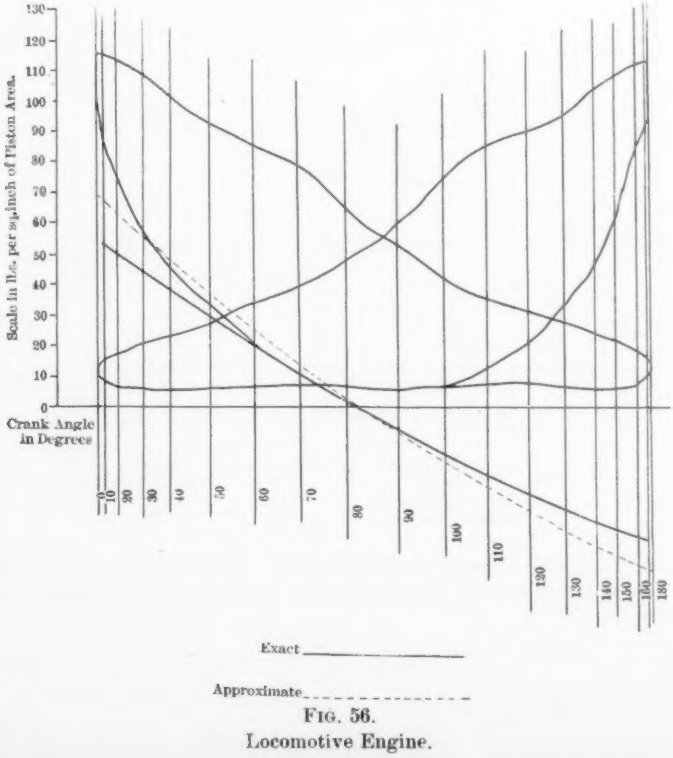


FIG. 56.
Locomotive Engine.

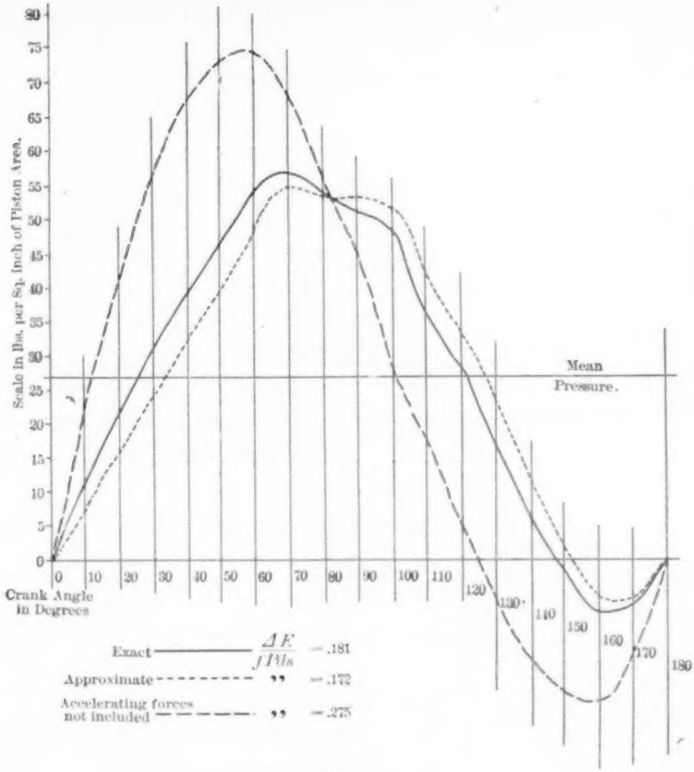


FIG. 57.
Locomotive Engine.—Tangential Effort Acting on Crank-Pin. Friction and Gravity Not Included.

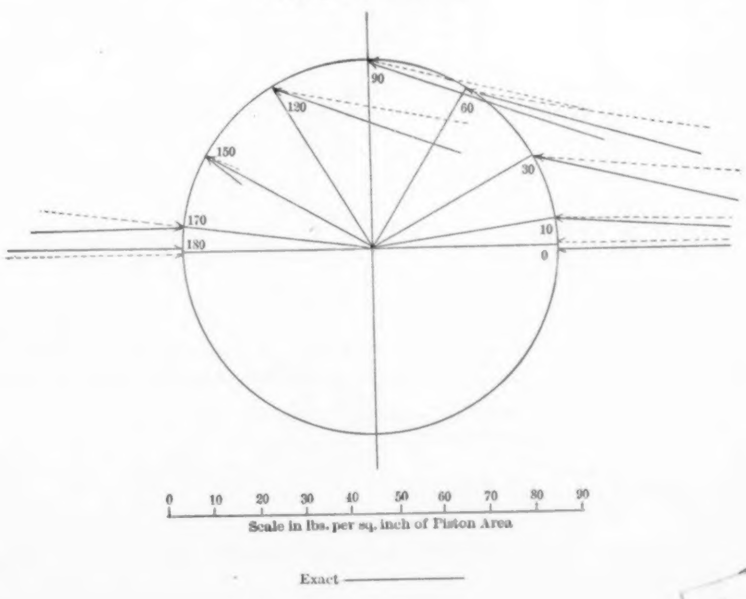


FIG. 58.
Locomotive Engine.—Pressures Acting on Crank-Pin.

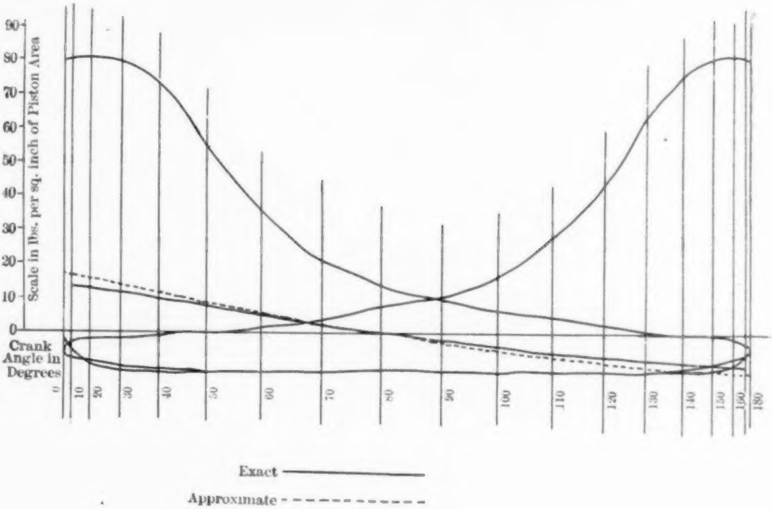


FIG. 59.
Corliss Engine.

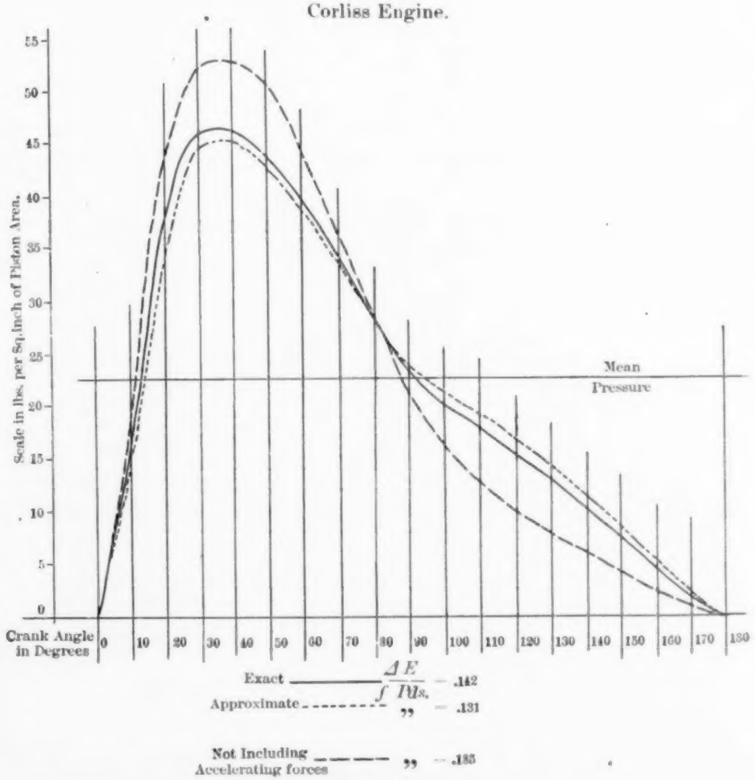


FIG. 60.
Corliss Engine.—Tangential Effort Acting on Crank-Pin. Friction and Gravity not Included.

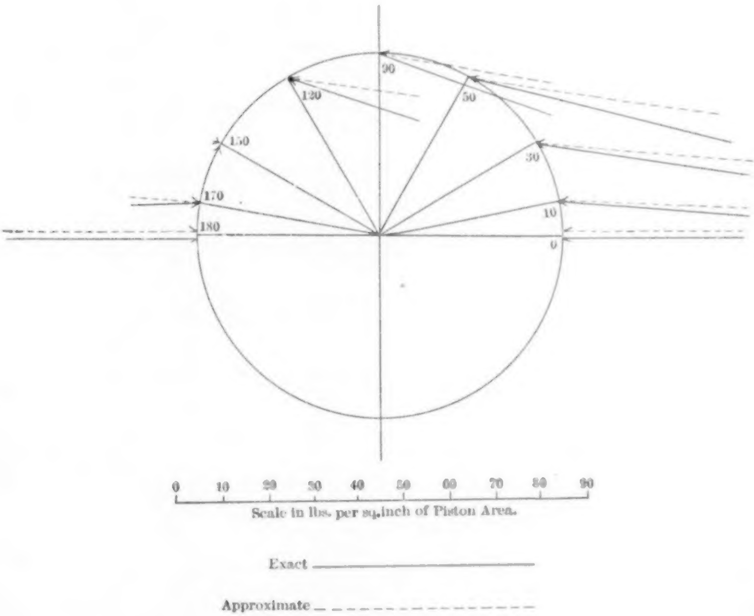


FIG. 61.
Corliss Engine.—Pressures Acting on Crank-Pin.

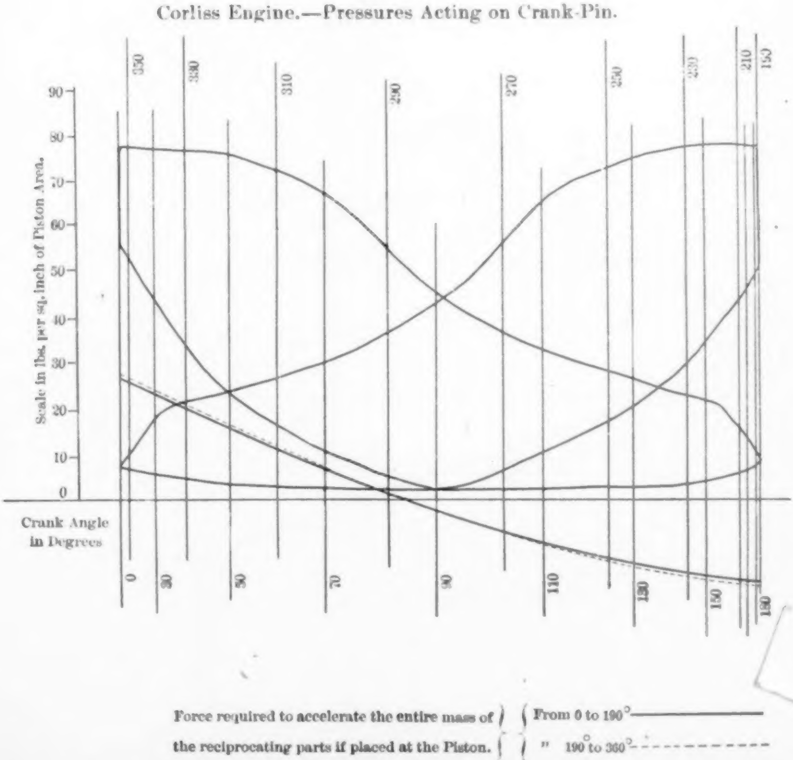


FIG. 62.
Westinghouse Engine.

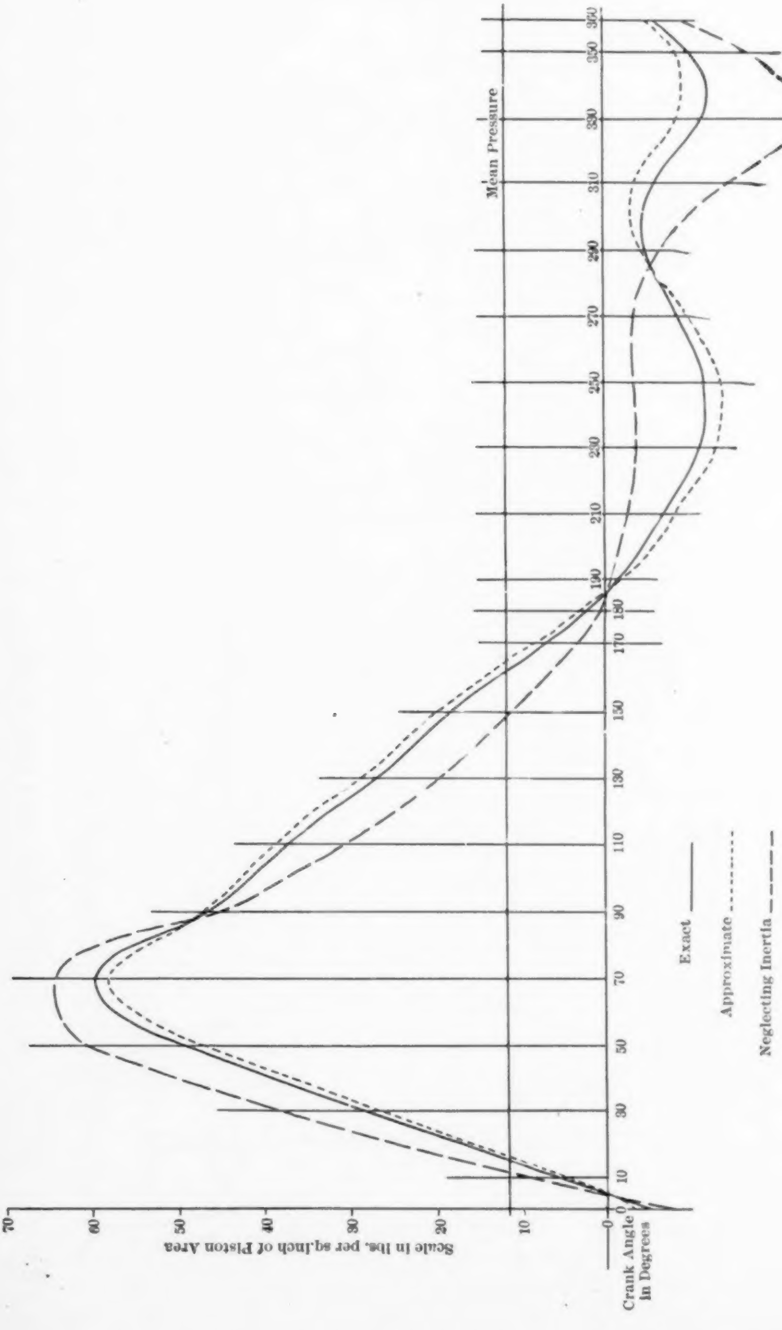


FIG. 63.

Westinghouse Engine. Tangential Effort Acting on single Crank-Pin.

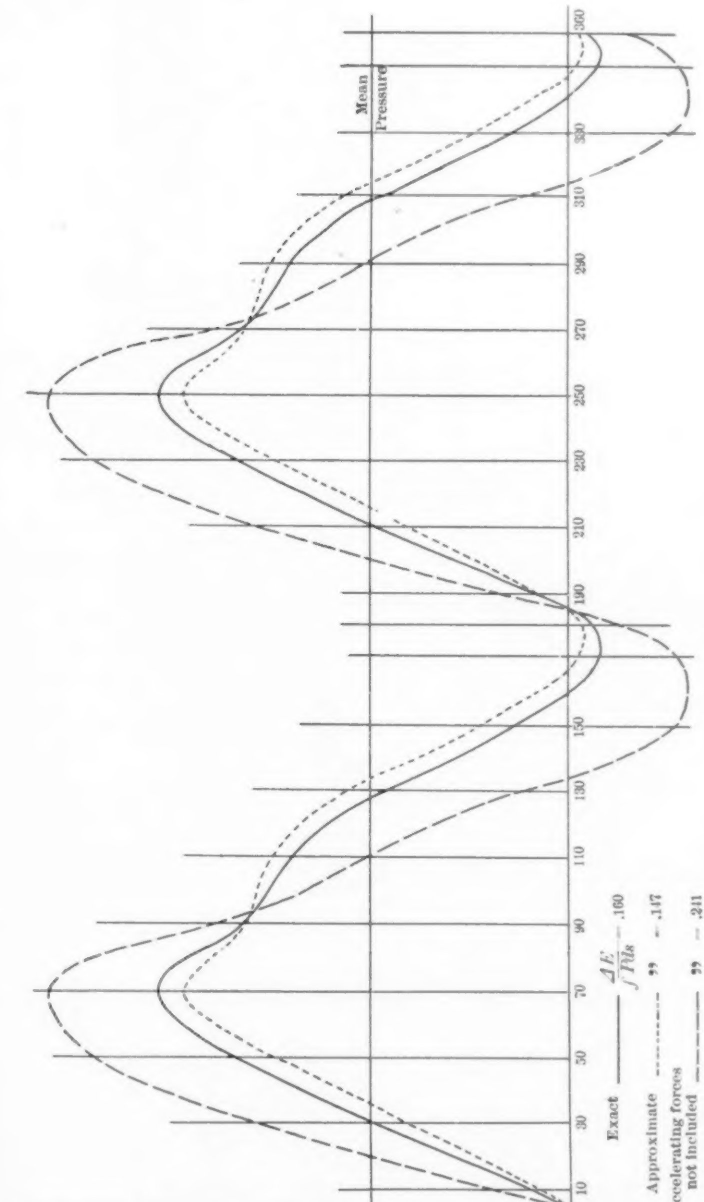


FIG. 64.
Westinghouse Engine.—Sum of the Tangential Efforts Acting on the two Crank-Pins.

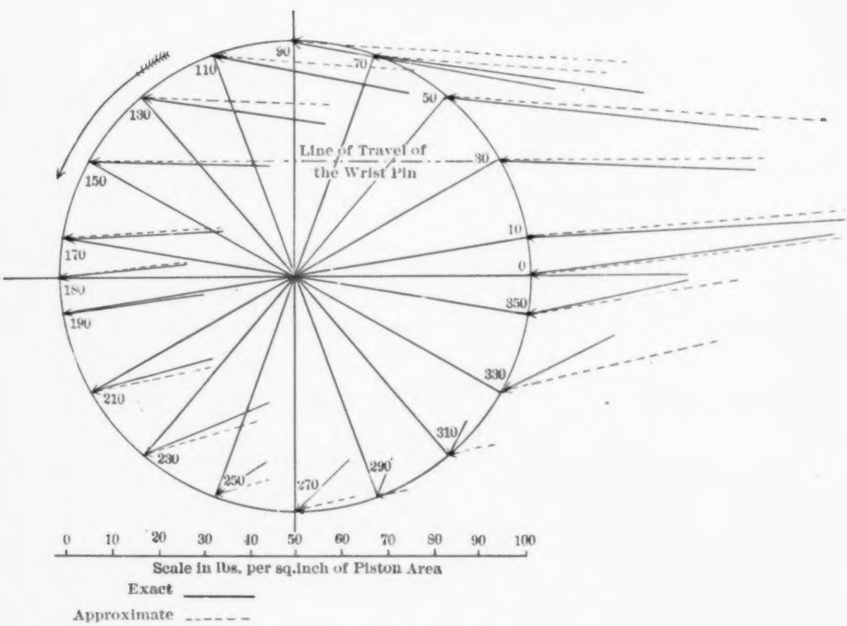
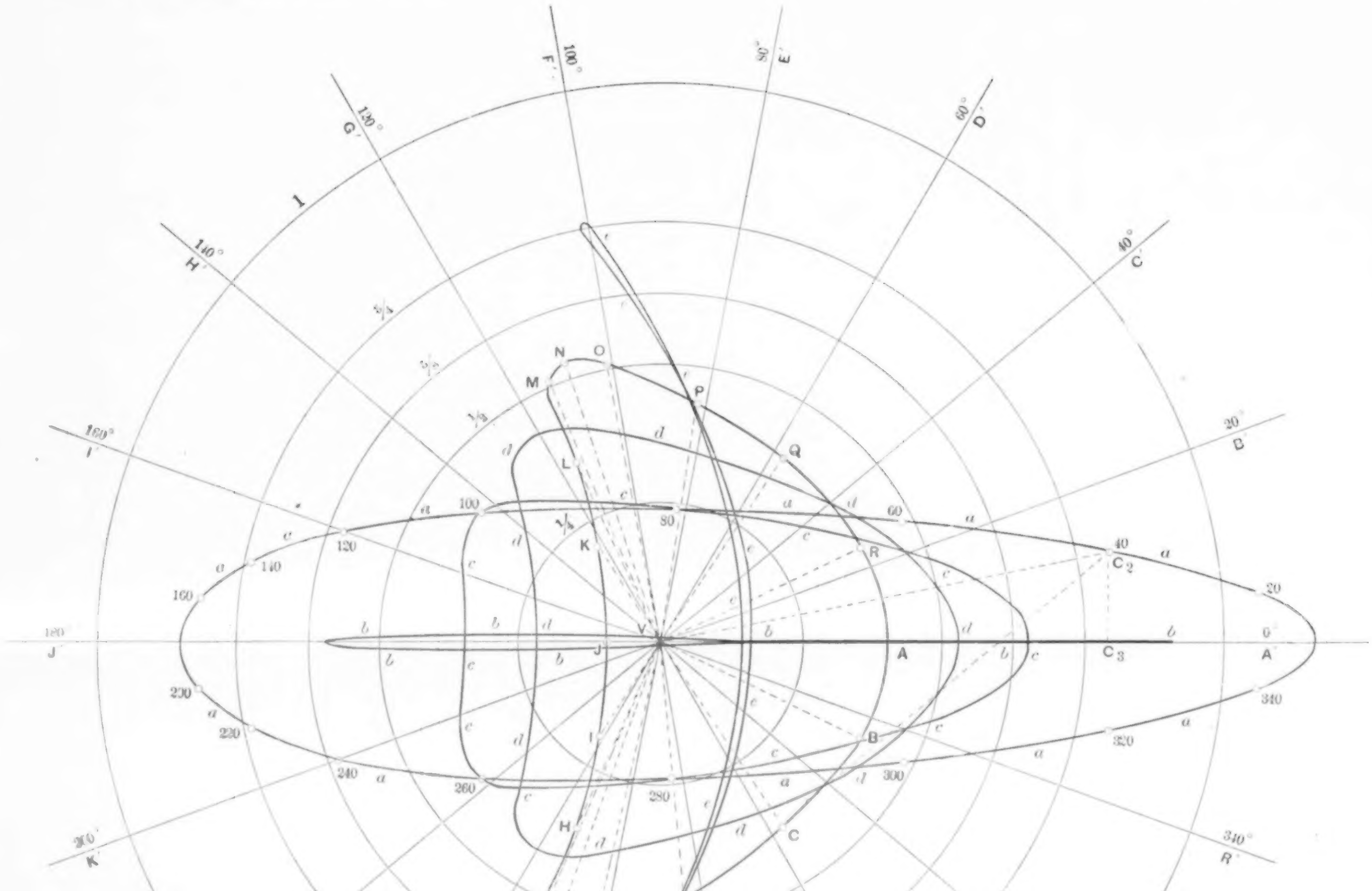


FIG. 65.
Westinghouse Engine.—Pressures Acting on Crank-Pin.


100°
F'
80°
E'
60°
D'
40°
C'
20°
B'
0°
A'
340°
R'
200°
K'
180°
J'
160°
I'
140°
H'
120°
G'
I
A
B
C
C2
C3
H
I
J
K
L
M
N
O
P
Q
R
V
20
40
60
80
100
120
140
160
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220
240
260
280
300
320
340

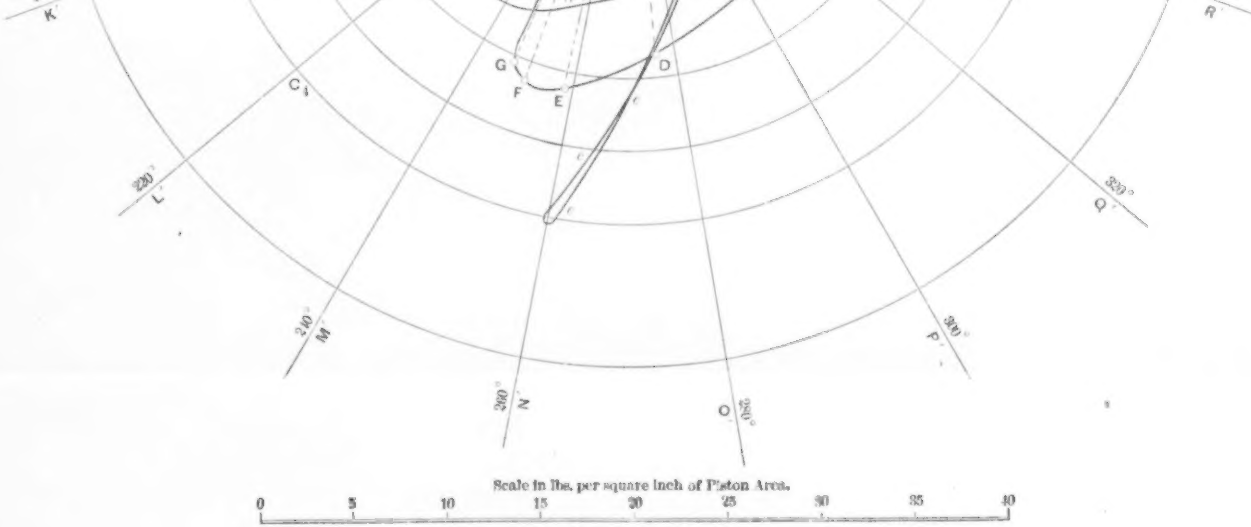

Fig. 66.

a, a, a, a, curve drawn through the extremities of the lines representing the shaking forces, assuming the engine to have no counterweight.
b, b, b, b, curve if a counterweight is employed equal to $\frac{1}{4}$ the mass of the reciprocating parts.
c, c, c, c, curve for counterweight equal to $\frac{1}{2}$ the mass of the reciprocating parts.
d, d, d, d, curve for counterweight equal to $\frac{5}{8}$ the mass of the reciprocating parts.
A, B, C, D, etc., curve for counterweight equal to $\frac{3}{4}$ the mass of the reciprocating parts.
e, e, e, e, curve for counterweight equal to the entire mass of the reciprocating parts.
Bore, 10″. Stroke, 12″. Revolutions per minute, 300. Cylinder at right of diagram.

APPENDIX.

FORMULÆ AND METHODS EMPLOYED IN DERIVING THE SAME.

FIG. 67 represents the main lines of an engine referred to the horizontal axis, OX, coinciding with the line of travel of the wrist-pin, O being the extreme point of its travel, and OY the vertical

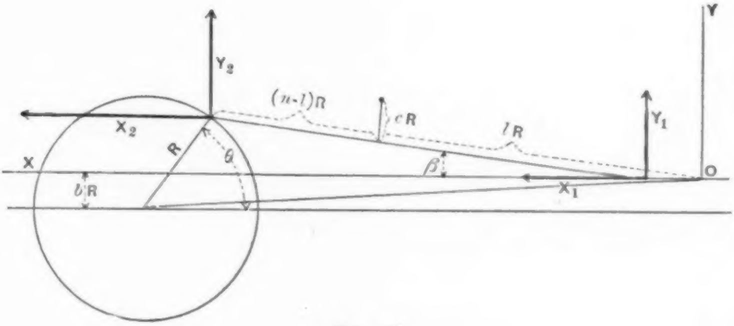

FIG. 67.

axis, the plus direction of the accelerating forces being indicated by arrow heads.

Let R = radius of the crank circle;
nR = length of connecting-rod;
lR = distance from the wrist-pin to the foot of the perpendicular let fall from the centre of gravity of the connecting-rod to a line joining the centres of the wrist and crank-pins;
cR = distance from the centre of gravity of the rod to the line joining the centres of the wrist and crank-pins;
kR = radius of gyration of the rod about its centre of gravity;
bR = distance from the centre of the crank shaft to the line of motion of the wrist-pin;
θ = crank angle measured from its position when parallel to the centre line of the cylinder;

β = connecting-rod angle;

δ = angle made by tipping the engine up about the crank shaft;

W_1 and M = weight and mass of the piston, piston-rod, and cross-head;

W_2 and m = weight and mass of the connecting-rod;

τ = angular velocity of crank shaft;

F_1 = accelerating force for piston, piston-rod, and cross-head;

X_1, X_2, Y_1, and Y_2 = horizontal and vertical accelerating forces acting in the direction marked in Fig. A;

Pa = pressure of steam on the piston;

G' = pressure of guides against the cross-head, acting at the centre of the wrist-pin at the angle $90° + \varphi'$;

φ' = angle of friction at cross-head guides;

φ_1 and φ_2 = angles of friction at the wrist and crank-pins respectively;

G = component of G' perpendicular to the line of motion of the wrist-pin;

P_w and P_c = forces exerted by the connecting-rod upon the wrist and crank-pins respectively;

T = tangential component of the force P_c acting at the crank-pin;

N = radial component of the force P_c acting at the crank-pin;

X_t and Y_t = horizontal and vertical components of the forces tending to shake the bed of the engine;

P_p = force that, when subtracted, the force P_a of the steam will give a result that may be compounded as if no inertia were present, in order to obtain the true value of T or N;

C = component of the weight of the piston, piston-rod, and cross-head, that acts in the direction of the centre line of the cylinder;

D and E = portions of the weight of the connecting-rod borne respectively by the wrist and crank-pins;

H = friction of piston and piston-rod;

r_w and r_c = radii of the wrist and crank-pins respectively.

The forces that are altered by the introduction of friction are marked by the subscript $_f$ when the equation includes this effect.

The only extra notation for friction which is not given above is T_f', which is the force T_f reduced to the centre of the crank-pin.

In the approximate formulæ P_p is the force required to accelerate the entire mass of the reciprocating parts, if it be assumed to move with the piston; this is the only accelerating force involved in this method, the rest of the notation being the same as for the exact.

Z is a quantity introduced in order to simplify the equations for the accelerating forces; its value will be given both for the general equations and those that are used in the simplified case of an engine in which the line of travel of the wrist-pin passes through the centre of the crank shaft.

FORMULÆ, WITH ALL THE GENERAL CONDITIONS OF THE PROBLEM INCLUDED.

$$F_1 = m\, r^2\, R\, \{\cos\theta + Z\}.$$

$$Y_1 = -\, m\, r^2\, R \left\{ \left(\frac{n-l}{n} + \frac{c\,(\sin\theta - b)}{n\sqrt{n^2 - (\sin\theta - b)^2}}\right)\left(\frac{l}{n}\sin\theta + \frac{c}{n} Z\right) - \frac{k^2}{n}\sin\theta \right\}.$$

$$Y_2 = -\, m\, r^2\, R \left\{ \left(\frac{l}{n} - \frac{c\,(\sin\theta - b)}{n\sqrt{n^2 - (\sin\theta - b)^2}}\right)\left(\frac{l}{n}\sin\theta + \frac{c}{n} Z\right) + \frac{k^2}{n}\sin\theta \right\}$$

$$X_1 = m\, r^2\, R \left\{ \left(\frac{n-l}{n} - \frac{c\sqrt{n^2 - (\sin\theta - b)^2}}{n\,(\sin\theta - b)}\right)\left(\cos\theta + \frac{n-l}{n} Z + \frac{c}{n}\sin\theta\right) + \frac{k^2}{n^2} Z \right\}$$

$$X_2 = m\, r^2\, R \left\{ \left(\frac{l}{n} + \frac{c\sqrt{n^2 - (\sin\theta - b)^2}}{n\,(\sin\theta - b)}\right)\left(\cos\theta + \frac{n-l}{n} Z + \frac{c}{n}\sin\theta\right) - \frac{k^2}{n^2} Z \right\}$$

$$G_f = \frac{1}{1 + \tan\varphi' \tan\beta}\Big[\big\{ P_a + C - F_1 - H + D\sin\delta - (A+B)\sin\beta - X_1 \big\} \tan\beta + Y_1 + D\cos\delta - (A+B)\cos\beta \Big]$$

$$P_{wf} = \sqrt{\{(P_a + C - F_1 - H - G_f \tan\varphi')^2 + G_f^2\}}.$$

$$T_f = \{P_a + C - F_1 - H - G_f \tan\varphi' + D\sin\delta - (A + B)\sin\beta - X_1\}$$
$$\sec\beta\sin(\theta + \beta) - Y_2\cos\theta - X_2\sin\theta - (A + B)\cos(\theta + \beta)$$
$$- E\cos(\theta + \delta).$$

$$N_f = \{P_a + C - F_1 - H - G\tan\varphi' + D\sin\delta - (A + B)\sin\beta - X_1\}$$
$$\sec\beta\cos(\theta + \beta) + Y_2\sin\theta - X_2\cos\theta + (A + B)\sin(\theta + \beta)$$
$$+ E\sin(\theta + \delta).$$

$$P_{cf} = \sqrt{T_f^2 + N_f^2}.$$
$$T'_f = T_f - B_n.$$
$$X_t = -(F_1 + X_1 + X_2).$$
$$Y_t = -(Y_1 + Y_2).$$

In which

$$Z = \frac{n^2\cos^2\theta}{\{n^2 - (\sin\theta - b)^2\}^{\frac{3}{2}}} - \frac{\sin\theta(\sin\theta - b)}{\sqrt{n^2 - (\sin\theta - b)^2}}.$$
$$A = \frac{P_{wf}\, r_w \sin\varphi_1}{n\,R}.$$
$$B = \frac{P_{cf}\, r_c \sin\varphi_2}{n\,R}.$$
$$C = W_1\sin\delta.$$
$$D = \frac{n - l + c\tan(\beta - \delta)}{n} W_2.$$
$$E = \frac{l - c\tan(\beta - \delta)}{n} W_2.$$

FORCES WHEN FRICTION AND GRAVITY ARE NEGLECTED.

The accelerating forces will be the same as when all the conditions are included, and those that are obtained by combining these with the force due to the steam, reduce to the following:

$$G = (P_a^1 - F_1 - X_1)\tan\beta + Y_1.$$
$$P_w = \sqrt{\{(P_a - F_1)^2 + G^2\}}.$$
$$T = (P_a - F_1 - X_1)\sec\beta\sin(\theta + \beta) - Y_2\cos\theta - X_2\sin\theta$$
$$= (P_a - P_p)\sec\beta\sin(\theta + \beta).$$
$$N = (P_a - F_1 - X_1)\sec\beta\cos(\theta + \beta) + Y_2\sin\theta - X_2\cos\theta.$$
$$P_p = F + X_1 + (Y_2\cos\theta + X_2\sin\theta) \div \sec\beta\sin(\theta + \beta).$$
$$P_c = \sqrt{T^2 + N^2}.$$

If the rod is symmetrical about its centre line and the path of motion of the wrist-pin passes through the centre of the crank shaft, the equations for the accelerating forces reduce to the following:

$$Y_1 = m\,\tau^2 R\,\frac{nl - l^2 - k^2}{n^2}\sin\theta.$$

$$Y_2 = -\,m\,\tau^2 R\,\frac{l^2 + k^2}{n^2}\sin\theta.$$

$$X_1 = m\,\tau^2 R\left\{\frac{n-l}{n}\cos\theta + \frac{(n-l)^2 + k^2}{n^2}Z\right\}.$$

$$X_2 = m\,\tau^2 R\left\{\frac{l}{n}\cos\theta - \frac{l^2 - ln + k^2}{n^2}Z\right\}.$$

$$Z = \frac{n^2\cos^2\theta - n^2\sin^2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}.$$

FORMULÆ USED IN APPROXIMATE METHOD.

$$P_p = (M + m)\,\tau^2 R\,(\cos\theta + Z).$$
$$T = (P_a - P_p)\sec\beta\sin(\theta + \beta).$$
$$N = (P_a - P_p)\sec\beta\cos(\theta + \beta).$$
$$P_c = (P_a - P_p)\sec\beta.$$

FORMULÆ WHEN THE ACCELERATING FORCES ARE NOT INCLUDED.

$$T = P_a\sec\beta\sin(\theta + \beta).$$
$$P_c = P_a\sec\beta.$$

ABSTRACTS OF THE METHODS EMPLOYED IN DEVELOPING THESE EQUATIONS.

The acceleration of the mass of the piston, piston-rod, and cross-head, produced by the force F_1, is determined by finding the space travelled by the piston for any crank angle θ, and obtaining the second differential co-efficient of this, with regard to the time t required to turn through the arc θ, on the assumption that the angular velocity τ is constant.

The space travelled by the piston from its position when on the dead centre, at which it is farthest from the crank, is:

$$s = \sqrt{(R + n\,R)^2 - b^2R^2} - R\left(\cos\theta + \sqrt{n^2 - (\sin\theta - b)^2}\right).$$

As dt is the time to turn through the arc $d\theta$, we have $\frac{d\theta}{dt} = \tau$.

These give for the acceleration:

$$\frac{d^2s}{dt^2} = \tau^2 R \left\{ \cos\theta - \frac{\sin\theta\,(\sin\theta - b)}{\sqrt{n^2 - (\sin\theta - b)^2}} + \frac{n^2\cos^2\theta}{\{n^2 - (\sin\theta - b)^2\}^{\frac{3}{2}}} \right\},$$

from which

$$F_1 = M\frac{d^2s}{dt^2} = M\tau^2 R\,(\cos\theta + Z).$$

The general equations for the forces required to accelerate the mass of the connecting-rod have been derived by two methods:

1st. By determining the forces required at the centre of gravity to translate the rod, and introducing a pair of equal and opposite forces to produce its angular acceleration about this same point; and,

2d. By assuming the mass of the rod to be divided into two equal portions concentrated at points in a line passing through the centre of gravity and lying in its plane of motion, the distances from the centre of gravity to each of the masses being equal to the radius of gyration of the rod about the same point, and determining the forces that will be required to produce acceleration when the mass is so divided. These forces will be the same as those required to accelerate the actual rod.

FIRST METHOD OF DERIVING CONNECTING-ROD FORCES.

The X and Y components of the translative force at the centre of gravity are:

$$F_x = m\,\tau^2 R \left\{ \cos\theta + \frac{n-l}{n} Z + \frac{c}{n}\sin\theta \right\},$$

and

$$F_y = -\,m\,\tau^2 R \left\{ \frac{l}{n}\sin\theta + \frac{c}{n} Z \right\}.$$

The moment required to produce the angular acceleration is

$$M_r = -\,m\,\tau^2 R^2 k^2 \left\{ \frac{\sin\theta}{\sqrt{n^2 - (\sin\theta - b)^2}} - \frac{\cos^2\theta\,(\sin\theta - b)}{\{n^2 - (\sin\theta - b)^2\}^{\frac{3}{2}}} \right\}.$$

F_x and F_y are divided between the wrist and crank-pins in such

a manner that no angular acceleration will be produced by the components, and the moment, M_r, is made to act by placing equal and opposite components at the ends of the rod. The sum of the components of the translative and rotative components at the two ends of the rod give us the values of X_1, X_2, Y_1, and Y_2, as already indicated.

SECOND METHOD OF DERIVING CONNECTING-ROD FORCES.

The X and Y translative forces for the mass nearest the wrist-pin are:

$$F_{dx} = \tfrac{1}{2}\, m\, \tau^2 R \left\{ \cos\theta + \frac{n - l + k}{n} Z + \frac{c \sin\theta}{n} \right\},$$

and

$$F_{dy} = -\tfrac{1}{2}\, m\, \tau^2 R \left\{ \frac{l - k}{n} \sin\theta + \frac{c}{n} Z \right\}.$$

Similarly, for the mass nearest to the wrist-pin, we have:

$$F_{ex} = \tfrac{1}{2}\, m\, \tau^2 R \left\{ \cos\theta + \frac{n - l - k}{n} Z + \frac{c \sin\theta}{n} \right\},$$

and

$$F_{ey} = -\tfrac{1}{2}\, m\, \tau^2 R \left\{ \frac{l + k}{n} \sin\theta + \frac{c}{n} Z \right\}.$$

Each of these rectangular components of the accelerating forces for the two masses is divided between the wrist and crank-pins, and the sum of the components at the latter point gives the accelerating forces X_1, X_2, Y_1, and Y_2, as before.

MODIFICATIONS OF THE EQUATIONS BY THE INTRODUCTION OF FORCES THAT ARE PRODUCED BY FRICTION AND GRAVITY.

It has been demonstrated in a paper prepared by Prof. J. B. Webb and the writer,* published a short time ago in the *Annals of Mathematics*, that the effect of friction at the connecting-rod bearings may be determined by introducing into the equilibrium polygons two forces, A and B, the values of which have already been given, at right angles to the centre line of the rod, the sum of A and B being applied at each of the pins.

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* See *Annals of Mathematics*, Dec., 1888.

DISCUSSION.

Prof. J. E. Denton.—Assuming that we have sufficient means of determining the forces of acceleration, etc., with every possible degree of accuracy, it may now be asked, to what extent is it desirable or necessary to take account of such forces in engine design? As is well known, the pioneer in this field of study was Mr. Charles T. Porter. He applied the principle of heavy reciprocating parts to his first Allen 12 × 24 engine, built at Whitworth's, and ran it at 200 revolutions per minute at the Paris Exhibition, at a time when such a speed was almost unknown outside of locomotive practice. Mr. Porter states that the engine ran as smoothly as a wheel, and attributes it to the principle of heavy reciprocating parts.

Mr. Arthur Rigg, in his treatise on the steam-engine, states that the weight of the reciprocating parts of this engine was 470 lbs., and that the force to accelerate this weight was roundly equal to

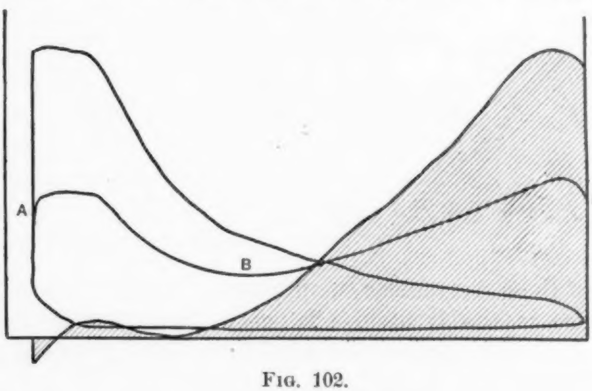

FIG. 102.

the boiler pressure, so that for a cut-off as per Fig. 102 the pressure on the crank was as per the shaded area; that is, the pressure was practically zero for one-fifth of the stroke, and then *uniformly* increased. Mr. Rigg considers this distribution a misuse of the Porter principle, and my attention has been called to this fact as proof that the quiet running of Porter's Paris engine was due to workmanship, and not at all to any aid from inertia. Mr. Rigg holds that if the reciprocating parts had been such as to absorb about half the boiler pressure, thus making the crank pressure about as per the line *ABC*, a much better action would have been had.

Now, when the Southwork Foundry built the Porter-Allen en-

gines, all that workmanship could do to make engines run smoothly was certainly done for them. The reciprocating parts in engines of about 14 inches diameter were of such weight as to absorb about half the boiler pressure, giving the crank-pin about the pressure represented by the line *ABC*. Yet these engines gave trouble from pounding on the centres, and were only made quiet by the adoption of so much compressive action that the steam was cushioned to nearly boiler pressure, thus making the distribution the same as in a high-speed locomotive, which has long been known to owe its smoothness of action to this amount of cushion. Since then high-speed engines generally have adopted the "cushion" to about boiler pressure, and the weight of the parts is naturally such as to make the force of acceleration about equal to half boiler pressure. Now, under these circumstances the pressure on the crank is zero just before the end of the stroke, and increases *uniformly from zero up to half boiler pressure;* practically just as did the pressure on Porter's Paris engine.

On the other hand, if there is no cushion and the pressure is nearly uniform as per line *ABC*, there is a *sudden* change in crank pressure from half boiler pressure in one direction to half boiler in the other direction, and this makes the pounding. I therefore believe that Mr. Porter's Paris engine *did* derive its smoothness from the Porter principle.

Regarding the value of balancing pressures, or avoiding irregular action, I would relate the following incident:

A 7 × 14-inch engine, running 215 revolutions, had its port at one end stopped up, and was run single acting for about nine weeks. At the end of that time the fly-wheel, four feet diameter and weighing 1,200 lbs., had sheared off the key holding it to the main shaft. The engine, to the eye, ran as steadily as when running double acting.

Again, a $5\frac{1}{2}$ × 7-inch engine ran compounded with a 7 × 14-inch at 160 revolutions, the connection being by belt through a 2-inch line of shafting. Although the revolutions were as nearly the same as could be counted, yet the slight difference of harmony in the running of the two engines sheared off the pin of a Sellers coupling on the connecting shaft. My point is, that if the irregularities of action present in these cases can do these things, it is worth while to design engines in the light of the fullest possible knowledge of the principles of acceleration laid down so exhaustively in the present paper.

Prof. R. H. Thurston.—This paper is one which, in a certain way, is of extraordinary value, and I think the preliminary work should also appear in our transactions in full, if there is no reason to the contrary. While results are often of extreme value, yet they represent only a part of the work. If any one hereafter should desire to revise the work, he would find it a great convenience to have it all in the transactions in one place, and not be compelled to look for parts of it in other places and at other dates. It is not, of course, likely that any one will ever find it necessary to make much use of such equations, but it constitutes a standard of the practice which it is very important to have constructed, and equally important to have on record in accessible form.

Mr. Jesse M. Smith.—I had occasion a few years ago to design a high-speed engine which has since been put into practice, and I went pretty thoroughly into the question of the acceleration of the moving parts; and I found, as Mr. Jacobus has stated, that it is impossible to put any counter-weight into the crank which will counterbalance all of the acceleration forces. Those forces can be divided into two sets—those which act horizontally and those which act vertically. It is perfectly feasible to counterbalance one set or the other, but it is impossible to attain both results at the same time. I adopted the process of finding what was necessary to make the horizontal balance, and what was necessary to make the vertical balance, and then according to circumstances used one or the other. If the engine was horizontal, in which case the accelerating forces were largely horizontal, I thought it best to put in a counterbalance which was almost equal to the horizontal accelerating forces, and letting the weight of the frame and the weight of the foundation take up the vertical action. Of course if the engine were to be a vertical one, resting on a heavy foundation—which is absolutely necessary in that case, more so than for a horizontal engine—the accelerating forces which are horizontal would tend to tip the engine over, and those which are vertical would act directly up and down to lift the foundation off of its seat and set it back again. I found that a very smooth-running horizontal engine could be made by making the weight of all the reciprocating parts equal to the weight of the counterbalance—that is, supposing that the centre of gravity of the counterbalance is the same distance from the centre of the shaft that the crank-pin is from the centre. In a horizontal engine the vertical components of the accelerating forces are rather small compared with the horizontal, and I found it hardly necessary

to compensate for them. I agree very thoroughly with what Prof. Denton has said—that Mr. Porter was right in putting counterbalance into his engine for the purpose of making it run smoothly. I cannot quite follow Mr. Porter as far as he went in putting in a "reciprocating fly-wheel," as he called it, but the idea is a good one, and as long as we have to have weight to the reciprocating parts it is necessary to put a counterbalance which will overcome their effect, if we expect to have an engine which will not drive itself off the foundation.

Mr. Geo. M. Bond.—I might say in this relation, without going into the reasons for it, that the company with which I am connected has a Straight Line engine of about 20 H.P., which was placed on the floor of the first story of a large new brick building, in a part used as a pattern room. Although the engine was carefully balanced for lateral motion (horizontally), it was found impossible to secure it vertically without shaking the floor considerably, at the speed at which it was designed to run. Its speed was about 240 revolutions per minute, I should think, and as it was only placed there temporarily it was soon after moved to the basement, and by giving it the proper weight of foundation the vertical disturbing forces were resisted. The weight of the foundation and the strength of the bolts which secured it overcame the difficulty perfectly.

Mr. Scott A. Smith.—I should like to call Professor Denton's attention to the fact that in the Porter-Allen engine, as now constructed by the Southwork Foundry, they connect the exhaust valves much lower down on the link, and they get, therefore, much more compression than they formerly did. The use of the heavy piston by Mr. Porter was, I suppose, to prevent the pounding which might occur in the engine. With the peculiar valve gear of the Porter-Allen engine, to get a short cut-off the opening of the steam valves has to take place by a lead, particularly if there is any "lost motion" between the eccentric and the valves. Consequently if there is not compression up to about the admission line there is apt to be a pound; but a very heavy piston takes up the force of the blow from the sudden admission of the incoming steam. As I said, connecting the exhaust valves lower down on the link, they now get on the Porter-Allen engine much more compression than formerly, thus gradually taking up all lost motion, if any exist in the reciprocating parts.

Prof. Denton.—I would like to have Dr. Sellers tell the history

of the change of the compression on the Porter-Allen engine. I think he can tell all about it.

Dr. Sellers.—Unfortunately, I cannot fully reply to Prof. Denton's request. It is a matter which has been done in our works, but I have not the facts at hand now to make useful remarks on the subject. The early Porter-Allen engines put on the Pennsylvania Road and those to run the electric lights of William Sellers & Company made a great deal of noise, due to the attempt to avoid any cushion and thus increase efficiency. But many of the changes to which you allude now were made during my illness, or about the beginning of my enforced retirement from active work.

Prof. Denton.—Was it not suggested from your works?

Dr. Sellers.—Oh, yes; the alterations required to stop the noise by means of cushioning the steam emanated from the works of Wm. Sellers & Co.

Mr. Jesse M. Smith.—It strikes me that acceleration and compression are quite different things. It is necessary to put in counterweight for the purpose of preventing the horizontal forces from moving the engine on its foundation, but that has no effect upon what lost motion there may be in the connecting-rod. What is sometimes called the Porter-Allen knock in an engine I think is entirely due to the lack of compression, and that the compression is put in simply for the purpose of taking up gradually the lost motion at the cross-head and at the crank. I think that the putting in of the compression is really what silenced the knock in the Porter-Allen engine; it was no change in the arrangement of the counter-weights, or of the weight of the reciprocating parts.

Mr. Scott A. Smith.—I had no reference to the counter-weight or balancing of the crank; it was simply the heavy piston to take up the blow from the steam which exists in a Porter-Allen engine as formerly made, from the fact that to get a short cut-off it is necessary to open the valve before the piston gets to the end of the stroke, as just explained.

Mr. Jesse M. Smith.—That is, the piston had to be stopped by lead—by letting in live steam rather than compressing the steam already there.

Mr. Scott A. Smith.—That is exactly the idea.

Prof. Denton.—I regard this principle of acceleration, and that stopping of the noise of the Porter-Allen engine, as two different things. I remember distinctly an engine in which the rapping occurred and they tried to reduce it as much as they could, and it

was never satisfactorily done until this matter of lowering the connection of the link was made, thus getting the extra cushion. You could put your ear to the engine afterward, and every pulsation which formerly you could hear a hundred feet off you could just distinguish. But acceleration reduces the total pressure on the crank-pin at the beginning of the stroke, and distributes it more evenly and tends to make the rotation more uniform, and would not produce these effects I speak of, like shearing off the pin by the jerky action of the fly-wheel, which cannot be seen.

Mr. Scott A. Smith.—In the Corliss engine Mr. Corliss never, so far as my experience went, found it necessary to use any compression. In fact you could not get compression on his engine using one eccentric. Compression was not necessary, from the fact of the non-opening of the steam valve with such suddenness as exists on the Porter-Allen engine, as explained. I think this is a subject of a good deal of interest, and to be followed out to its end.

The President.—The chair would like to second the suggestion made by Professor Thurston, that in a paper having so much value and original matter it is very desirable that the whole of the mathematical work should be included, and for one reason additional to those mentioned by Dr. Thurston—namely, that, to an increasing extent, the papers and transactions of the Society are gaining currency in a direction that we are all glad of—namely, among engineers in other countries; and that while the record to which he refers may be in another series of transactions accessible to American engineers, it might not also be accessible to foreign engineers. I suggest, therefore, that it would be better to have the whole of the mathematics of the subject incorporated with the paper.

Prof. Jacobus.—I thank the gentlemen for their expression of approval of the paper, and shall be glad to present the complete mathematical analysis for publication by this Society.*

With reference to that part of the discussion in which compression and counterbalancing have been compared, I agree in the main with what has been said. Compression has an internal effect upon the engine, while the effect of counterbalancing is external. The effect of compression is to introduce two equal and opposite forces in the engine, one acting on the piston and the other on the cylinder head; and two such forces have no external effect—that is, they do not affect the motion of the engine as a whole and thus

*** Appendix II., which has been added since the meeting, contains the complete mathematical analysis.**

produce a shake. Counterbalancing, or the introduction of an additional mass into any one of the moving parts, must, however, affect the motion of all the others, and will therefore alter the shaking of the bed. Smoothness of running, if it means absence of internal pounding, is affected by compression, while if it includes non-shaking of the bed it is affected by the counter-weight.

I agree with what Prof. Denton has said in regard to the effect of ununiform rotation.

[NOTE.—By reason of a delay in the issue of the *Annals of Mathematics* for February, 1890, the mathematical basis of the paper could not be appended to it, as suggested, without entailing inconvenient delay in issuing the *Transactions*. This part of the paper has therefore been made an appendix by itself to the second part of this volume.—*Secretary*.]

CCCLXXVIII.

APPENDIX II.

TO THE

REPORT OF THE COMMITTEE ON STANDARD TESTS AND METHODS OF TESTING,

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE.—This Appendix forms part of the Report of a Committee of the Society, and was appended to it when the Report was provisionally submitted at the New York meeting in November, 1889. The Report itself is not yet ready for publication, but the Committee has decided to publish this Appendix in advance of the Report, to give it circulation among engineers and to benefit by their suggestions upon it in prefacing the final Report to the Society.

G. C. HENNING,

Reporter for the Committee.

RESOLUTIONS OF THE CONFERENCES

HELD AT

MUNICH, SEPTEMBER 22–24, 1884,

AND

DRESDEN, SEPTEMBER 20–21, 1886,

RELATIVE TO

UNIFORM METHODS OF PROCEDURE

IN

TESTING BUILDING AND STRUCTURAL MATERIALS,

TO DETERMINE THEIR MECHANICAL PROPERTIES; COLLATED BY ORDER OF THE CONFERENCE AT DRESDEN BY THE EDITORIAL COMMITTEE,

J. BAUSCHINGER,
Professor at the Technical High School at Munich.

FR. BERGER,
Superintendent of Buildings, Vienna.

G. EBERMEJER,
Counsel to the Board of Directors of the Royal Bavarian Railways.

DR. F. HARTIG,
Professor at the Technical High School, Dresden.

L. TETMAJER,
Professor at the Technical High School, Zurich.

MUNICH, 1887.

TRANSLATED BY GUS C. HENNING, M.E.

Reporter to Special Committee on Standard Tests, A. S. M. E.

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INTRODUCTION.

It is hardly necessary at this day to furnish proofs that tests of the resistance of materials can only be compared when they have been made according to a uniform standard method.

Even the standard rules for furnishing and testing Portland cements, which were established by the association of German cement manufacturers as long ago as 1876, resulted from the perception that such agreement was necessary. Likewise the specifications for furnishing axles, tyres and rails of Bessemer metal and steel, which were proposed in 1879 by the general convention of the association of railway managers of Germany, advising their adoption by the managers of the association.* But these first steps to establish standard methods of testing emanated from quarters which from their nature included only technical specialists; on the one hand, purely manufacturers, and on the other, purely consumers; and although, in establishing these standards or norms, it was sought to make them as unprejudiced as possible, particular interests predominated, and this was urged against them on behalf of the opposite party.

Moreover, these deliberations related to one group of materials alone. For this reason it was certainly an apposite undertaking when, in the autumn of 1884, professional men of the most varied technical callings assembled at Munich, to attempt to formulate uniform methods of testing of all of the more important materials of construction.

Eventually, a number of important questions were agreed upon; certain others were referred to a standing committee, which, later, considered them primarily by letter and discussed them orally,† later on, at two meetings held at Munich 21st and 22d of September,

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* The properties of iron and steel, VIII. Supplementary volume of the publications of the progress of railway construction. Wiesbaden, 1880.

† The detailed report giving the transactions of the conference at Munich, and the constitution of its standing committee are contained in the XIVth. part of "Mittheilungen aus dem mechanisch-technischen Laboratorium der technischen Hochschule München." (Munich, Theo. Ackermann.)

1885; finally, the results and conclusions were presented at the second general conference held at Dresden, September 20 and 21, 1886. The latter concurred in most of the conclusions presented; but there still remained a number of unsolved problems which again were entrusted to a second standing committee, whose duty it should be to discuss them and then present their findings to a third conference to be convened in the fall of 1888 at Berlin. Meanwhile, the conclusions arrived at, as well as the questions still undetermined were to be collated in a convenient form in the shape of a pamphlet to be as widely distributed as possible.

This task was confided by the Dresden conference to the committee mentioned on the title page.

In presenting this pamphlet as the result of our work we cannot resist expressing the wish that, in order to make it fulfil its mission as nearly as possible, it may keep alive the interest in the aim of the conferences, and to call it into existence in such quarters which heretofore have held aloof.

To further this object we deem it necessary that we state that each conference is a voluntary assembly of men assembled for a free exchange of opinions upon the best methods or systems of testing particular materials, which are to serve certain purposes. If resolutions are voted at these conferences, they can only decide which method of procedure is considered the best by the *majority* of those present.

According to the first resolution of the first conference "the discussions are to be free, and the resolutions not binding." There are no obstacles to a subsequent conference resuming a question which had been decided by a previous one, in order to be again discussed and voted upon. Methods of testing cannot remain the same; they *must be developed in accordance with our increasing knowledge of the properties of materials used in construction;* with the improvements made in their production; with the adoption of new materials for the same, etc., etc., etc.

It, therefore, becomes necessary that those who take interest in testing materials, from a scientific standpoint, or because they are manufacturers or consumers, assemble from time to time, in order to deliberate afresh, exchange opinions, and to mutually learn and teach, and then as a result of their deliberations agree upon methods of testing which are by them or at the least by the majority considered the most satisfactory, to be adopted for the time being.

In the following collation the resolutions already adopted are printed in ordinary type, their causation in small type, while those questions referred to the standing committee, as well as the unsolved problems, are in italics:

1. GENERAL DETERMINATIONS.

1. Every *testing machine* must be so arranged that it can be readily and accurately tested or rated.

The construction thereof must be such that when properly operated all *impact* due to loading is avoided.

This property appertains as well to hydraulic as to screw machines.

For practical purposes an arrangement for making the machine work automatically is not necessary.

2. Good *shackles* or *holders* must be so constructed that the tension or pressure be distributed as uniformly as possible over the cross section of the test piece.

Therefore the requirements:

α. for Compression Tests:

a. Free and easiest possible motion of one of the tables or supports in all directions.

b. The surfaces on which the pressure is applied must be as nearly as possible parallel, and for this end they must be planed or turned when the material permits it.

β. for Tension Tests:

Free and easy movement of the test piece for adjustment at the beginning of test. These conditions are fulfilled according to experience.

For *round rods*, by the ball bearing, most satisfactorily such with undivided spherical shell.

For *flat bars*, pinhole with pin, and with one hole and bolt at each end only, or milled ends and corresponding wedges.

Pummer's Holder, a kind of universal joint, with two knife edges at right angles to one another, is recommended for *vertical* machines. "*Its use in horizontal machines, however, as well as the use of others which may yet be proposed or designed, are referred to the second standing committee for further study.*"

The use of serrated wedges, *i. e.*, of such steel wedges which force themselves into the test piece, are to be discarded.*

3. Any one *standard* form of machine for testing, in daily practice, cannot be recommended; it is, however, to be mentioned that a number of the well-known apparatuses answer their particular purposes more or less perfectly.

4. To the results published by the experimenter, are always to be annexed such short notes upon the machines and methods employed as are necessary in judging of the value of the results obtained.

5. To the results of tests obtained are to be added, whenever possible, in addition to the source of the test piece, a microscopic or chemical examination, or both, as well as notes on the process of manufacture of the test piece, and other physical, chemical, or technical characteristics.

Such amplification of the results of tests will rarely be possible in *routine testing;* it is, however, highly desirable, and never to be neglected in scientific researches.

6. *The influence of time on the results of tests of materials, according to Fischer and Hartig, cannot be doubted; but the construction of an apparatus showing the relation between the results obtained and the rate of testing was connected with many difficulties. Prof. Hartig was successful only shortly before the assembling of this conference; he will continue his investigations and report them to the next conference to be convened.*

7. The materials are to be tested for those qualities or properties for which they are adopted in construction.

The quality of a material itself is the resultant of all of its mechanical properties. As long as the interdependence of these several properties is unknown, so that the existence of several might lead to the determination of others—and we are still far from that point—so long the observation of a few of the properties will not be sufficient to judge of the applicability of the materials for different purposes, and they must be tested for those properties for which they are adopted in construction.†

8. Such materials which are acted upon by *percussion* or shock in structures are to be tested by *drop tests*, in order to determine their character.

9. The drop tests are to be made by means of a *standard* "*drop.*" For such the following essentials were adopted:

* For description and sketches of the above recommended holders, see Part XIV. of the "Mittheilungen aus dem mech. techn. Laboratorium der techn. Hochschule München," pp. 287-290, and Table II.

† The detailed discussion of this decision is given in Part XIV. of the "Mittheilungen," etc., pp. 156-160.

a. It was not considered advisable to adopt the complete design of a standard "drop," but accurate requirements are to be exacted for those details which can influence the results of tests.

It was thought necessary to bear in mind those existing "drops," which for the most part conform in their essential parts to the following requirements.

b. Every "drop" is to be standardized.

It is not impossible that drops, which had been constructed with all possible care, give erroneous results, affected by unavoidable influences.

c. With due regard to the existing drops, and as the results are dependent only upon the work done, drop weights of 500 kilos. (1102.25 lbs.) and 600 kilos. (1322.82 lbs.) were considered applicable; the 500 kilos. (1102.25 lbs.) weight is, however, strongly recommended for adoption.

d. The weight may be made of cast iron, cast or wrought steel; the shape is to be such as to place its centre of gravity as low as possible.

The face of the weight is to be made of forged steel, and to be placed accurately symmetrical to the vertical axis through the centre of gravity of the weight by dovetail and wedges. Special marks are to indicate the accuracy in this respect.

The vertical line through the centre of gravity of the weight must be central to the guides. Special marks on the anvil are to indicate this centre line.

e. The length of the guides on the weight shall be greater than twice the clear width between the guides.

The guides are to be made of metal, *i. e.*, rails, and to be so constructed that the weight has very little play. Greasing the guides with graphite is recommended.

f. Detachment or Release must not produce oscillation of the weight between the guides, and, therefore, the escapement must be easily and freely controllable. The point of suspension must be truly above the centre of gravity of the weight. Between escapement and the weight there is to be a short movable link; *i. e.*, chain or rope. The method officially employed in Russia is particularly recommended, as shown in Fig. 105.

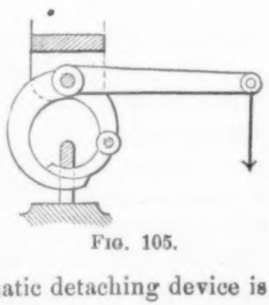

FIG. 105.

g. With constant height of fall, an automatic detaching device is recommended.

h. The bearings for the test piece are to be rigidly attached to the scaffold or frame, *i. e.*, to be wedged fast.

The bearing blocks should always be as nearly as possible in one piece with the scaffold or frame.

i. The weight of the frame and anvil block shall be at least ten times that of the drop weight.

Inasmuch as tests with the ballistic drop and with an ordinary drop which had a weight of anvil of only four times that of the weight, gave sufficiently corresponding results (see Kick "on the law of proportional resistances") it may be confidently assumed that with a tenfold weight of frame and anvil block the results will invariably be comparable.

k. The foundation shall be inelastic and consist of masonry, the magnitude of which should be determined by the locality and subsoil.

l. The surface struck should always be level; therefore, proper shoes or bearing blocks are to be attached when testing rails, axles, tyres, springs, etc., etc., to provide an upper flat surface. These blocks are to be as light as possible.

A uniform drop weight with flat surface of contact is recommended in order to simplify its possible reconstruction as a standard drop and to require but a single calibration according to § *o* and § *p*.

m. In regard to the shape of bearing blocks and caps, no directions are as yet given in view of the still missing results of experiments; it is, however, recommended that a description of the exact shapes used be given, whenever reports of tests are published, or when certificates of test results are issued.

n. Drops up to 6 m. (19.68 ft.) high ought to be considered more reliable than those of greater height, and it is, therefore, recommended that 6 m. (19.68 ft.) be not exceeded. For heavier tests a drop weight of 1000 kilos. (2204.7 lbs.) ought to be used.

Drops up to 6 m. (19.68 ft.) high can be more readily erected under cover and with greater reliability; they are less liable to alteration of detail than those that are larger.

o. The work of impact is determined by the effective weight and the distance through which it falls. The total mass of the weight dropped is to be so rectified that the effective weight is a convenient number, *i. e.*, 500 kilos. (1101.25 lbs.).

Comparable results are only obtainable when the loss due to friction is considered or eliminated.

p. The following methods can be adopted to determine the effective weight of the drop:

α. A spring balance of proper force is attached between the

weight and the lifting rope, and the indication read *while* the weight is slowly *descending;* thus the dead weight *less* the friction is obtained. While raising the weight, the dead weight *plus* the friction is found.

p. The weight of the drop is determined by the effect produced by the blow due to a given drop upon a standard plug of best stay-bolt copper placed exactly central, and which has an exact shape and weight which are still to be determined.

q. Such standard copper plugs are also to be used to compare different drops and to standardize them.

The Standing Committee shall advocate, conjointly with the Directing Committee of the royal technical testing institutions at Berlin, that the several establishments under its direction shall be intrusted with the preparation and control of the standard copper plugs to be used in the calibration of drops, and these plugs are to be accompanied with a table giving the changes of form produced by the application of work due to blows.

r. Drops which have an absorption of work due to friction of more than 2% of that produced by shock are to be discarded.

For *governing drop tests* only the most perfect drops are to be used, while on the other hand all those poorly constructed or entertained are to be avoided.

For the *method* of *testing*, the following principles are laid down:

s. The standard drop is intended principally to test *whole* or *finished pieces*, such as rails, axles, tyres, springs, etc., etc.

Tests of specially prepared pieces are of the greatest value, but instructions for the construction of a special small drop or the establishment of rules for the execution of such tests are not considered advisable, inasmuch as such tests are primarily of a scientific character. The construction of such smaller apparatus for public laboratories and the execution of guiding tests or investigations are to be recommended to the proper authorities.

t. The guides are always to be in a truly vertical position, and this as well as correct position of the weight in the guides is always to be ascertained. The point of impact vertically below the centre of gravity of the weight is to be carefully marked on the anvil and is to be verified before each test, in order to place that point of the test piece to be struck exactly below the centre of gravity of the weight, and the test piece must be so placed that no tilting or displacement is produced by the impact.

u. Inasmuch as the work done in a drop within very wide limits is dependent only upon the product of effective weight by the

height of fall, but not upon the magnitude of either of the factors, it is recommended to adopt the metre-ton as the measure of work, and for the work to be done only such quantities which are multiples of five hundred; the divisions on the scale must then be divided into half metres instead of metres.

v. It is recommended to adopt sliding scales in order to be able to place the zero point always on a level with the top of the test piece.

w. At present it is considered sufficient if deflections of test pieces 1.0 m. (3.28 ft.) to 1.5 m. (4.92 ft.), between bearings be determined to 1 mm. (0.0394 in.).

x. With the view of making results of tests comparable, it is recommended when publishing results of tests to annotate all details of procedure during the test (*i. e.* the order of blows, whether continuous or intermittent, whether the test piece was reversed or not), as well as all details of behavior of test pieces.

y. The Standing Committee shall examine new propositions for drops and drop tests, and particularly gather and collate published experiments and researches with the view of establishing a standard method of test.

II. TESTING OF WROUGHT IRON * AND STEEL.

A. Rails.

1. Rails, for reasons of safety of traffic, and in accordance with the decisions under I. No. 8, must be subjected to the *drop test*, by means of proper technical arrangements (standard drop, see I. No. 9).

2. When further knowledge of the quality of the material is desirable, the *tension test* is to be made.

3. Finally, rails should be subjected to *bending test* (by static load) and in two ways: one to determine the "yield point" by permanent set; the other to determine the amount of *permanent deflection* under excessive loads beyond the elastic limit.

That tension tests *alone* were not sufficient for testing rails was the universal opinion of the Munich Conference. The contradictions of the results of such tests as emphasized by *Tetmajer*, and as demonstrated on Finnish, Swiss, and French railways, plainly gave evidence to the contrary. If, on the one hand, these con-

* This name is adopted in contradistinction to cast iron, and is intended to include weldable iron and low steels.

tradictions are the result of the fact that the rails are subjected to shock, and on the other hand, the tension test merely indicates the character of a small part of the rail section, it certainly seems advisable to lay more stress upon the drop and bending tests, but emphasizing the application of *proper* apparatus, for the former, especially the standard drop. As these latter, however, do not give as much information about the character of the material as tension tests, particularly when accompanied by chemical analysis, they must be still resorted to, as long as such information is desirable. With our comparatively limited experience thus far gained in that respect, this will no doubt have to be done for some time to come. Comparative results of both kinds of tests of the same material, such as are now being executed at Berlin, will have to decide which procedure is to be preferred, or whether both are to be used hereafter conjointly.

4. For testing rails by tension, the test piece is to be cut in *rectangular form* from the extreme elements.

As a concomitant reason for the unreliability of tension tests of rails, is to be mentioned the fact that the test pieces heretofore were turned specimens taken from the middle of the head, and did not at all test the parts most subject to wear and tear, such as the head and foot; and furthermore, that the injurious bubbles of silicious gases in metal, poured very hot, are always found near the surface of the ingots. On the other hand, it is to be remarked, that pure steel solidified by the admixture of manganese contains a porous central zone in the ingots, which may affect the tension test, but in no way deteriorates the rail, and may thus be the cause of erroneous conclusions.*

5. *The formulation of proper methods of determining wear of rails and tyres, is synonymous with the research of proper methods of determining wear in general, and seems such a difficult problem that its solution will certainly still require considerable time, and is for the present turned over to the Standing Committee.*

6. The investigations relative to the effect of different *kinds of tyres* upon the *wear* of rails must be classified as a special field of operation for railway administrations.

B. Axles.

1. Axles are to be subjected to the *drop test* at the middle as well as the ends.

2. *Tension tests* of the same are to be resorted to when additional knowledge of the material is desirable.

3. Axles need not be subjected to the *bending test.*

C. Tyres.

1. Tyres are to be subjected the same as rails and axles to the *drop tests.*

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* See pp. 42–53, Part III. of "Mittheilungen."

2. *Tension tests* are to be resorted to where additional information is desired as above.

3. *Hammer test* is *not* necessary.

D. Multiple or piece tests.

1. *It is desirable to obtain as much information as possible about multiple tests (tests of every piece of a lot of material) in order to be able to formulate a standard method of procedure.*

2. *In the construction of drops and testing machines the possibility of making multiple tests is to be kept in view.*

3. *Not only axles are to be kept in view but also all building and structural pieces of iron and steel.*

The multiple test, *i. e.*, a method which is as rapid a test-process as possible, such as a blow given to each piece of a delivery to judge of the quality, without, however, doing any injury, is certainly as appropriate to assure a higher degree of certainty, as the one now generally in vogue of testing merely a percentage of the delivery or a number of pieces of a load. It has long ago been adopted for springs, chains, pipes, boilers, cylinders, etc.; as in Austria, where the multiple test has been frequently applied to axles with very good results. On the other hand, the fact must not be lost sight of, that its application is combined with considerable difficulty for buyers as well as manufacturers, which might, however, be overcome by the introduction of a proper method of testing. For this reason the question of multiple tests was referred to the standing committee.

E. Wrought iron for bridge construction.

1. Wrought iron for bridge construction shall be subjected to the *tension tests* as well as,

2. The *bending test*, and this by a static load applied by mechanical means, the material to be tested hot as well as cold, and bent around a stud uniformly of 25 mm. (0.985 inch) diameter.

The conditions to be fulfilled by this mechanical device are to be discussed by the standing committee.

3. The flattening or spreading as well as fracture test need not be resorted to, as the two tests above indicated sufficiently characterize the material.

F. Low steels for bridge construction.

1. Low steels as well as wrought iron are to be subjected to the *tension test*, as well as

2. The *bending test*, by a static load applied slowly by mechanical means, the material being tested hot and cold, and bent around a stud uniformly 25 mm. diam.

Welding tests and determinations of hardness are not to be made. The former, because no welding is done in bridge construction, and is indeed to be avoided;

the latter for the same reason, and because the character of the material is sufficiently determined by the tension and bending tests.

For special detail of bridges (bearing plates, etc.), hardness is indeed required, but special requirements are then prescribed for them; moreover, greater strength carries with it greater hardness.

G. Wrought iron for boiler work.

1. For the three shapes used in boilers made of wrought iron the following tests are to be made:

a. For *plates.*

1. Tension test.
2. Bending test.
3. Forging and punching tests.

b. For angle irons.

1. Tension test.
2. Bending test.
3. Forging and punching tests.

c. For rivet rods.

1. Tension test.
2. Bending and forging tests.

2. Angle irons need not be tested for their welding qualities, but this is desirable.

3. Red shortness, which could be established by the welding test, is determined by the bending test with hot specimens.

H. Low steels for boiler construction.

Low steel (Bessemer, Martin and Thomas materials), when taking the place of wrought iron in boiler construction (stationary locomotive and marine boilers, etc.), shall, in order to conform to standard methods, be tested as follows:

1. By *tension test.*

2. By the *bending test, cold* and at a *red heat.* The edges are to be trimmed, and for plates above 6 mm. (.236 inch) thickness the specimens are to be bent to a given angle, around a stud uniformly 20 mm. (.985 inch) diameter, by the use of a slow-operating mechanical device.

By these tests the behavior of the material when actually worked, and its suitability for the work is determined. The stud uniformly 25 m. per .985 inch diameter is convenient, and harmonizes with most of the present specifications.

3. By the *quenching test.* The specimens with edges trimmed are to be uniformly heated throughout to a dull cherry red (about

550–650° C (1022–1202° F), and quenched in water of a temperature of 25° C (77° F), and then tested as stated under 2.

Experience has taught that the softness of low steels for boiler work is to be such that the hardness produced by quenching is so slight that it shall in no way prevent the material from being worked readily. Material having a tenacity of from 38–42 kg. per sq. mm. (50,000–60,000 lbs. per sq. in.), with 20% elongation is generally suitable; but still it is safer to determine this point by the above quenching test.

4. By the *forging test* (spreading test), which is always to be made at a red heat.

The punching test need not be made, as punching of low steel plates is to be avoided, on account of the peculiar incipient fractures produced.

5. Addendum: Those establishments which habitually use low steels are recommended to test the material as well for its *welding* properties.

The principal reasons against the general introduction of the welding test, are the difficulty of its execution, its dependence upon the degree of skill and experience of the workman, and also the circumstance that the material is weldable with some difficulty, but not with such certainty that a riveted joint is not, perhaps, preferable. The rolled and corrugated Fox boiler flues are, however, welded, as well as the ordinary flues, and low steel gas conduits, but they are all individually tested.

Weldable low steel, moreover, does not harden when quenched, or only very slightly, and is much less affected by temperature, but this is determined by the quenching test.

Tests of low steel plates after being annealed can be omitted; the following reasons make it appear not advisable: Such plates are not always annealed before use, principally on account of the cost. Only plates of small diameter are rolled or curved while hot; hand flanging is done by heating the plates locally little by little; corrugated or pitted heads are no longer annealed, as are flat heads to prevent distortion.

The presence of internal strain cannot be decided by this test.

It is difficult to determine the temperature at which the material is annealed, and this, as well as time, exert considerable influence. Determinations of temperature would make the tests very laborious. And, after all, material is always to be tested in the condition in which it is delivered.

In favor of testing the material after annealing, may be mentioned that comparable results of tests of plate can only be obtained after eliminating internal strains by annealing, *i. e.*, its natural condition; also, that the same material when rolled into plates of various thicknesses gives varying results; and finally, that comparisons of tests of the material as from the rolls and annealed discover defects, which are produced by careless manipulation during fabrication.

J. Wire.

Wire is to be subjected to the following tests:

1. *Tension test.*

2. *Winding test*, and this by mechanisms which exclude arbitrariness.

3. Bending test, by *repeatedly bending* the wire *forward* and *back*, by the use of mechanism, bending around a stud uniformly 5 mm. diam.

Establishing a stud of definite diameter precludes the abominable habit of clamping the wire in a vise with square jaws, over which the wire is bent forward and backward.

K. Wire-ropes.

Wire-ropes are to be tested by:

1. *Tension test.*

2. Test by *impact* and *concussion* longitudinally.

The bending test would only then be valuable when applied to determine durability, but that is, practically, very difficult and unnecessary, as each wire in the rope has already been tested for reverse bending.

L. Measurements during the Tension Test.

1. During tension tests the following observations are to be made:

a. Tenacity.

b. Contraction at the point of rupture.

c. Elongation after rupture (compare under *M*, No. 1).

d. Limit of elasticity or limit of proportional extensions.

Although the contraction is determined essentially by shape and dimensions of section, it is for that reason important to establish standard directions (see *M*).

Actually it would answer the purpose better to determine the elongation under the maximum load or at the beginning of the local contraction, but this point it is difficult to determine with large machines.

This elongation might, however, be determined with sufficient accuracy for practical purposes, from two observations which eliminate the local influences, one of which is the elongation after fracture of a length of 10 cm. and the other that of 20 cm.

2. It is recommended for the determination of the *diagram of work done* (resilience) to make as many individual or single observations, in case a special apparatus is not already used for recording this diagram.

3. When taking such diagrams it is of essential value to indicate the *time* (rate) of taking them.

4. When taking diagrams the following five points are to be determined with the greatest possible precision:

a. Elastic limit (limit of proportional extensions).
b. Yield-point.
c. Instant or point of contraction.
d. Maximum load (drop of beam).
e. Rupture.

5. The contents of the diagram of work of resistance are to be considered up to the point of rupture.

Only that part of the work done by the rod up to the point of contraction need be determined ; after that instant the part contracting is the one mainly doing work, and the latter is of no great value in most structural materials ; as the determination of the area of the diagram of work up to the point of rupture can produce no great error in the whole value, it may as yet be retained, as the determination of the instant of maximum load is very difficult. Furthermore, the separate consideration of that part of the diagram obtained after contraction commences is desirable, because possibly there may exist some relation between the extension of test piece after that point is reached and the work done to produce it.

M. Shape of Test Piece for Tension Test.

1. *Round rods* are to be prepared in four types, and all of the same length between points of measurement of 200 mm. (7.88 in.), standard length, but with diameters of 10, 15, 20, and 25 mm. (.394, .591, .788, .985 in.), according to requirements or possibility.

By standard length of 200 mm. (7.88 in.) is to be understood that the turned part of a specimen measures 220 mm. (8.668 in.) between shoulders, and all observations are made on the 200 mm. (7.88 in.) in the centre, leaving 10 mm. (.394 in.) at each end between gauge marks and shoulders.

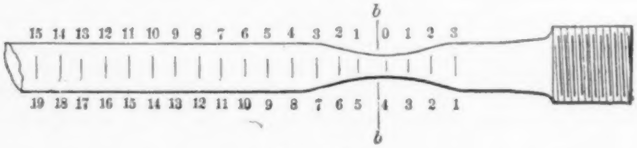

FIG. 107.

This working length is to be divided into parts of 10 mm. (.394 in.), and the elongation after rupture (see *L*, No. 1, *c*) is to be determined as follows : Supposing the rupture to occur as shown in Fig. 107, between divisions 4 and 5. Measurements can be made to the left from 1 to 5, or 1 to 10, adding 0-*b* and 1-*b* to each (using figures inscribed above the rod) accordingly as 10 or 20 cm. (.394 or .788 in.) was used as standard length. Toward the right, how-

ever, only 0-3 can be determined, and assuming that the elongation beyond the section contracted is identical, the value taken from 3-5 or from 3-10 accordingly as 10 mm. or 20 mm. are assumed as standard length, is to be added to obtain the total elongation of 10 mm. or 20 mm. In this way the measurements are so taken that the elongation is nearly the same as though the rupture had taken place in the centre of the specimen.

This dividing (marking) and measurement of elongation after rupture is always to be on two opposite sides of the test piece.

2. The standard length of *flat rods* is the same as for round rods, 200 mm. (7.88 in.), this length meaning the same thing in both cases. The dividing or marking, as well as the determination of elongation, are to be the same in the two cases.

For the reason given under *L*, No. 1, it is recommended to extend the determination of elongation to 10 cm. (.394 in.) as also to 20 cm. (.788 in.).

3. In such cases where width and thickness of test pieces of *rectangular section* can be chosen at will, the width shall be 30 mm. (1.18 in.) and the thickness 10 mm. (.394 in.), but in all cases the section of 30 × 10 mm. is to be considered as the standard.

In place of the width of 50 mm. (1.97 in.), formerly customary, one of 30 mm. (1.18 in.) is to be substituted, with reference to the smaller testing machines, largely used in works, which generally can exert a force of but 50 tonnes (55.12 tons).

4. Where the thickness is fixed in plates, the width of test piece is to be 30 mm. (1.18 in.) up to a thickness of 24 mm. (.945 in.), and from 25 mm. (.985 in.) upward the thickness of plate is to be chosen as width of test piece, and this made 10 mm. (.394 in.) thick.

In order to retain the surface produced by the rolls in this case, the shoulders are to be cut out of the *width* of the plate.

Exceptionally, where smaller testing machines only are available, the limits of 16 mm. and 17 mm. (.630 and .670 ins.) may be taken instead of 24 mm. and 25 mm. (.945 and .985 in.).

5. Test pieces from flat bars, angles, tees, and I beams, as well as channels, etc., etc., are to be cut *lengthwise*, in the shape of *rectangular bars* of 200 mm. standard length, and not more than 30 mm. of width of section. When the thickness of bars, angles, channels, etc., etc., exceeds this, then specimens are to be cut *lengthwise*, so that the whole section is represented by the test pieces, by taking two or more lying parallel and side by side in the original bar, or as shown by Fig. 3, which represents a channel bar.

6. The surface in contact with the rolls must invariably remain on the test pieces.

7. *The respective proper authorities are to be requested to instruct the testing laboratories under their supervision to undertake, in behalf of the general public interest, the comparison of standard test pieces for tension tests as proposed by the conference.*

The comparison of the above round and rectangular types of test pieces for tension tests as to tenacity, elongation and contraction, is such a vast subject, that it cannot be completed by any one individual, and by private means. The co-operation of all official or governmental testing laboratories, and governmental financial assistance must be bespoken in behalf of this work.

III. TESTS OF CAST IRON.

1. *Test pieces* are to be in the shape of prismatic bars 110 c.m. (43 in.) long (100 c.m. standard length), and have a section of 3.0 c.m. (1.18 in.) square.

Larger sectional area would, however, be desirable for bending tests as well as for the facility of procuring specimens for tension tests, but in order to make them comparable with the fundamental experiments by Wachler, the above dimensions were retained.

2. These test pieces are to be cast *horizontally*. The pouring is to be done at two points simultaneously, the gates being one-third the distance from either end. When test pieces are cast differently, then the method of procedure must be exactly and carefully stated.

Wachler's test bars were cast on end (but it is not stated whether poured from below or above), but experience has taught that with some kinds of iron the metal chills too readily, when bottom poured, and experience in pouring from above is still lacking. The method of casting is, moreover, dependent upon the quality of the cast iron, upon the skill of the moulder and caster, etc.; therefore the above final injunction.

3. The pressure employed in the method of casting as required under No. 2 shall be equal to a column of cast iron 15 c.m. high.

4. Dry sand moulds are to be used.

5. During the test the points to be determined are as follows:

a. Resistance to flexure and work done during bending up to the point of rupture, using three test pieces.

b. The *tenacity* as determined from turned test pieces 20 mm. diam. and 200 mm. (.785 in. × 7.85 in.) standard length, cut from the two ends of the test piece broken by flexure, and two of these out of each of the three first tested.

c. Compression, determined from cubes 3 c.m. (1.18 in.) on each

side, cut from the pieces obtained by *a*, and two such out of each bar. Pressure to be applied in the direction of the axis of the original bar.

6. The bars for bending, and the cubes for compression test are to be tested as taken from the sand, with the scale.

7. Special cast pieces, such as bearing plates for bridges, water pipes, etc., etc., are to be submitted to special test in accordance with the uses to which the pieces are to be put.

IV. TESTS OF COPPER, BRONZE, AND OTHER METALS.

The establishment of uniform methods of testing copper, bronze and other metals required such manifold preparatory work, that no definite results have thus far been obtained, and the problem was turned over to the standing committee for further investigation.

V. TESTS OF WOOD.

1. Necessary parts of a technical opinion of woods are (as far as possible): Information of location of forest, the locality in which the tree grew from which the specimens were obtained, whether crowded in a dense forest or in an open spot; from which part of the tree the wood was cut; finally, statement of age and time of year when felled.

2. To form a technical opinion on woods at least three samples are necessary, on account of the great difference existing between single individuals and even pieces.

3. Of each test piece a description of external appearance is to be given as follows:

A. Of the *longitudinal cut*, or better still, surface as produced by splitting: making note of

a. Straight position of fibres or otherwise.

b. The presence or absence of knots or ingrown limb buts.

B. Appearance of *cross section:* giving

c. When examining leaf trees and all members belonging to the pine family:

α. The average radial width of rings in mm.;

β. The increase or decrease or the change of width of rings on one-half diameter;

γ. Circular or eccentric arrangement of rings, and

δ. In all pines the average proportion of spring and autumn growth of all the rings.

4. The specific weight of each test piece is to be determined in the condition of moisture in which it happens to be tested as well as after thorough drying in a temperature of from 101 to 105° C. (214–221° F.). The difference in moisture in these two conditions is also to be determined.

5. *Compression* and *transverse tests* are to be the means of determining the solidity or strength and resilience of woods.

a. For the compression test prismatic blocks are to be used, having a section of 10 cm. by 10 cm. (3.94 × 3.94 in.), and a length of 15 cm. (5.91 in.), with exactly concentric position and parallel bearing surfaces.

b. Transverse strength is to be determined from prismatic pieces 10 cm. by 10 cm. (3.94 × 3.94 in.), in cross sections, 160 cm. (63.0 in.) long, with 150 cm. (59.1 in.) c. to c. of bearings. The point of application of the force is to be protected against injury by a rider of 2 cm. (.788 in.) thickness, and 20 cm. (7.88 in.) length or other satisfactory means. Flexure is to be carried up to rupture. Rupture of single bundles of fibres or splitting in or separation of layers, is not to be considered as rupture.

c. The fibre strains at the time of rupture are to be determined from the usual formulæ for flexure, under the assumption that it is applicable up to the instant of rupture.

d. The determination of the resistance or quality of the wood is effected by means of the *work done* during *flexure* by the test piece of above dimensions, represented by the diagram of resistance up to the instant of maximum load.

6. In order to determine the resistance of whole trunks the separate sections of which show differences, the compression and bending tests as above are to be executed, and upon at least two pieces cut from the centre and two selected so that the extreme edges coincide with the surface of the trunk; from these the correct mean of the whole trunk is obtained.

7. The relative position of rings and direction of application of force in these tests is to be represented by a diagram. Sap-wood is to be strained in the direction of the radius of the rings with the heart on the concave and surface on convex side of curve produced by flexure.

VI. TESTING OF MATERIALS FOR SHIP CONSTRUCTION.

1. Materials used in ship construction, such as *wrought iron* in the shape of plates, bars, angles, shape and rivet iron, are to be tested by:

a. Tension test.

b. Bending test while *cold.*

c. Bending test while at a *red heat* or forge test.

2. Such as *low steels* and *cast steels*, in the shape of plates, bars, angles, shape and rivet iron, are to be tested by:

a. Tension test.

b. Bending test while *cold.*

c. Bending test while at a *red heat*, or forge test.

d. Hardening or *temper test* by flexure.

e. Those parties who habitually use low steels in ship construction are recommended to examine the material furnished for its welding properties.

The tests of resistance of the materials are to be made with standard test pieces, in the condition in which the material is furnished. The specimens for bending tests are to be obtained precisely as prescribed for boiler plates (see above under II., *G* and *H*).

The hardening and bending tests are to be made precisely as above, with the difference that the bars are to be bent by a slowly operating mechanical device, using, however, studs which will produce bending of the inner surface to a *radius proportional* to the *thickness of plate.*

VII. TESTING OF STONE.

A. Stone in General.

Stone is to be examined for its *resistance* to *boring* or *quarrying*, in accordance with uniform methods in the following manner:

1. *Trial methods.*

The trials for the determination of resistance to boring stone are to be made by,

a. A jumper or drill; or

b. By rotary boring machines.

For testing by jumper or drill is to be recommended an apparatus such as was used by mine surveyor Hausse, of Zankerrode, in his tests of the resistance of stone to boring, and which was

described in the "Deutsche Berg und Hüttenmännische Zeitung," 1882, Nos. 33 and 34. By such an one or a similar one, or else by a boring machine the work done in metre-kilograms required to bore a hole of given dimensions is determined.

2. *Determination of the proper preliminary work before the execution of the actual tests of strength.*

Because the least work done upon a stone of given quality with a given size of hole is largely dependent upon

a. The moment of drop of a jumper, or the force required to advance the borer as also its velocity of rotation, and

b. The shape and cutting angle of drill, or of the teeth of the boring tool, and

c. The number of blows for one revolution of drill, when drilling by percussion ; it is recommended to determine by preliminary trials, the most favorable combination of conditions under which the actual tests are to be carried out. As a starting point it may be stated, that for a drilled hole 25 mm. diameter, the most favorable moment of drop lies between 6 and 9 metre-kilograms, and that the pressure in boring machines varies from 30 to 130 atmospheres; also that in percussion drills, the most favorable angle of cutting edge lies between 70° and 110° *C*, with from 12 to 60 blows per one revolution of drill; and finally that with existing boring machine the most favorable diameter of bore varies from 40 mm. to 80 mm. (1.576—3.152 inches).

3. *Special standard procedure.*

After the most favorable method of working the particular variety of stone has been determined by the preliminary investigations, a diameter of hole of 25 mm. (.985 inch) shall be adopted as the invariable standard when using a *percussion drill* (jumper), this being about the approximate hole drilled by a single (one man) drill. However, in order to determine the effect on the work done, of the size of hole drilled, holes of larger diameter are to be drilled, and for this 35, 45 and 65 mm. diameter (1.379, 1.773 and 2.561 inches), corresponding to the average holes produced by two and three men or power drills, are recommended. When these larger diameters of holes are investigated by drilling, then the most favorable moment of drop (moment of impact) is to be determined, on the basis of the law of proportional resistances, starting with the most favorable moment of impact for drilling a hole 25 mm. (.985 inch) diameter which has been determined empirically beforehand.

For *boring experiments* no fixed diameter can be prescribed, as this is dependent upon the existing boring machine at hand, but it is recommended in such cases to adhere as closely as practicable to the diameters of drill holes of 45 and 65 mm. (1.773 × 2.561 inches).

4. *Additional tests.*

For the purpose of information it appears desirable that stone examined for resistance to boring be also investigated for compressive strength, elasticity and resistance to shearing.

5. *Test report of boring test.*

For the reports of results obtained the following blank form is to be filled up as indicated:

STANDARD REPORT BLANK FOR BORING TEST.

1.	Description of stone in its geological and mineralogical relations.
2.	Miner's classification (hard, very hard, or extremely hard).
3.	Texture (*i. e.*, coarse grained, fine grained, parallel, normal to or inclined to axis of drill hole.
4.	Specific gravity of the stone.
5.	Diameter of hole drilled.
6.	Diameter of hole and core when boring.
7.	Straight or curve edged drills.
8.	Angle of edge of drills.
9.	Number of blows per revolution of drill.
10.	Effective weight of drill.
11.	Mean effective drop of drill.
12.	Number of blows required to drill the depth of hole.
13.	Number and form of teeth of borer.
14.	Statement of pressure on and velocity of borer while boring.
15.	Actual or total depth of bore hole.
16.	Calculated or indicated work done during boring stated in metre-kilo grams per c.m. of hole bored. (When using a hollow borer the annulus of stone cut away is alone to be considered.)

B. Building Stones.

a. Natural Building Stone.

1. In addition to the petrographic and geologic description of the stone, its fracture and location of stratum from which the sample is obtained are to be clearly stated. Also the time of quarrying, as well as the deposit. When the quarry is very wet the sample is to be obtained in the dry season. Inasmuch as it is often difficult for the investigator to examine the correctness of minerological nomenclature given by the client, it is to be recommended to desist expressly from this examination unless the contrary is

requested, and to so record it in the test report; on the other hand, it is desirable to rectify palpable incorrectness of nomenclature of the stone, by a communication relative thereto to the client.

Similarly in a given case, the verification of statements relative to locality and stratum of quarry from which the sample has been obtained, may be also neglected, and the test report can be made to read about as follows: Stated to be obtained from the ——— quarries, found in the ——— bed or stratum.

2. It is to be recommended that the investigator make particular inquiries about the intended utilization of the stone by the client (whether as building or cutting stone, for sidewalks, for pavements, or for ballast or macadam), before commencing the tests, and then to execute them *in accordance* with *these*, and not the accidental wording of the order.

3. Stone, which is to be used as cut stone for sub or superstructure, is to be tested for *resistance* to *compression*, and in the shape of cubes, with planed bearing surfaces lying between pressure plates without the interposition of other substances; one of these plates must have free motion in all directions. The compressive strength is to be determined normal or parallel to the bed of the stone, or both, according to the utilization of the stone, and the tests in each direction are to be made in triplicate.

Test pieces should be selected as large as possible, according to the relative strength and the capacity of the testing machine, but for weaker stone a cube measuring 10 c.m. (3.94 inches) on each side is sufficient.

4. In addition to this, rectangular pieces are to be tested, from which the compression due to increasing loads is to be determined and the diagram of work constructed. Similarly tension and bending tests are to be made and plotted.

5. Test pieces shall be tested in a perfectly dry state, obtained by submitting them to a temperature of 30° C. (86° F.) until there is no further diminution of weight.

6. The specific gravity (weight of the unit of volume) of stone is invariably to be determined, and always after drying at 30° C. (86° F.).

7. The examination of *resistance to frost* is to be determined from samples of uniform size, inasmuch as the absorption of water and action of frost is directly proportional to the surface exposed, and with a view to the same test of cements this block is to be a cube of 7 c.m. (2.76 inches) length of edges. Smaller dimensions

are only permissible with very hard stones; although the frost-resisting qualities in these are rarely in doubt.

8. The *frost test* consists of:

a. The determination of the *compressive strength* of *saturated* stones, and its comparison with that of dried pieces.

b. The determination of *compressive strength* of the dried stone after having been frozen and thawed out twenty-five times, and its comparison with that of dried pieces not so treated.

c. The determination of the loss of weight of the stone after the twenty-fifth frost and thaw; special attention must be had to the loss of those particles which are detached by the *mechanical* action, and also those lost by solution in a definite quantity of water.

d. The examination of the frozen stone by use of a magnifying glass, to determine particularly whether fissures or scaling occurred.

9. For the frost test are to be used:

Six pieces for compression tests in dry condition, three normal and three parallel to the bed of the stone, provided these tests have not already been made (as under No. 3 above), in which it is permissible, on account of the law of proportions, to use test blocks larger than 7 c.m. (2.76 in.) cube.

Six test pieces in saturated condition, not frozen however; three tested normal to, and three parallel to bed.

Six test pieces for tests when frozen, three of which are to be tested normal to, and three parallel to bed of stone.

10. When making the freezing test the following details are to be observed:

a. During the absorption of water, the cubes are at first to be immersed but 2 c.m. (.77 in.) deep, and are to be lowered little by little until finally submerged.

b. For immersion distilled water is to be used at a temperature of from 15° *C.* (59° F.) to 20° *C.* (68° F.).

c. The saturated blocks are to be subjected to temperatures of from − 10°—15° *C.* (14° to 5° F.).

d. The blocks are to be subjected to the influence of such cold for four hours; and they are to be thus treated when completely saturated.

e. The blocks are to be thawed out in a *given quantity* of distilled water at from 59° F. —64° F.

11. An investigation of *weathering* qualities, stability under influences of atmospheric changes, can be neglected when the frost test has been made. However, the effects in this respect in nature

are to be carefully observed, and experience previously had in the utilization of such material is to be carefully collated, with special observation of

a. The effect of the Sun, in producing cracks and rupture of stones.

b. The effect of the Air, whether producing carbonic acid gas.

c. The effect of Rain and moisture in regard to partial solution and disintegration; and finally the effect of

d. Temperature.

β. *Artificial Building Stone.*

1. When testing bricks as found in a delivery, the least burnt are always to be selected for investigation.

2. Bricks are to be tested for *resistance* to *compression* in the shape of cubical pieces, formed by the superposition of two half bricks, which are to be united by a thin layer of mortar consisting of pure Portland cement, and the pressure surfaces are also to be made smooth by covering them with a thin coating of the same material. At least six pieces are to be tested.

3. The *specific gravity* of bricks is to be determined.

4. In order to control the uniformity of the material the *porosity* of the bricks is to be determined; for this purpose they are to be thoroughly dried and then submerged in water until saturated. Ten pieces are to be thoroughly dried upon an iron plate and weighed; then these bricks are to be immersed in water for twenty-four hours, in such a way that the water level stands at half the thickness; after this they are to be submerged for another twenty-four hours, then to be dried superficially and again weighed; thus the average quantity of water absorbed is determined. The porosity is always to be calculated by volume, though the per cent. of water absorbed is always to be stated in addition.

5. *Resistance* against *frost* is to be determined as follows:

a. Five of the bricks previously saturated by water are to be tested by compression;

b. The other five are put into a refrigerator which can produce a temperature of – 15° C. (5° F.) at least, and kept therein for four hours; then they are removed and thawed in water of a temperature of 20° C. (68° F.). Particles which might possibly become detached are to remain in the vessels in which the stone is thawed until the end of the operation. This process of freezing is repeated twenty-five times and the detached particles are dried and compared by weight with the original dry weight of stone. Particular atten-

tion, by using a magnifying glass, is to be given to the possible formation of cracks or laminations;

c. After freezing, the bricks are to be tested by compression. For this test they are dried, and the result obtained is to be compared with that of dry brick (see above, No. 2) not frozen;

d. Thus, freezing the bricks, does not give a knowledge of the absolute frost-resisting capacity; the value of the investigation is only relative, because by it can only be determined which stone can be most easily destroyed by the action of frost.

6. To test bricks for the *presence* of *soluble salts*, five are selected, and again those which are least burnt, and then such which have not yet been moistened. Of these again the interior parts only are used, for which reason the bricks are split in three directions, thus producing eight pieces of which the corners lying innermost in the brick are knocked off. These are then powdered, until all passes through a sieve of 900 meshes per square centimetre (= about 5,840 per sq. in.), from which the dust is again separated by a sieve of 4,900 meshes * per square c.m. (31,360 per sq. in.), and the remaining particles on the latter are examined. Twenty-five grammes are lixiviated in 250 cubic c.m. of distilled water, boiled for about one hour, however, replenishing the quantity evaporated, filtered and washed. The quantity of soluble salts present is then determined by boiling down the solution and bringing the residue to a red heat for a short time. The quantity of soluble salts present is to be given in per cent. of the original weight of stone.

The salts obtained are to be submitted to chemical analysis.

7. Determinations of the presence of *carbonate of lime, pyrites, mica, and similar substances* are to be made upon the unburnt clay primarily, for which purpose, too, unburnt bricks are to be furnished. These are soaked in water and the coarse particles are separated by passing the whole material through a sieve having 400 meshes per square c.m. (about one-third mm. opening). The sand thus obtained is to be examined by the magnifying glass and hydrochloric acid to determine its mineralogical composition. When impurities such as carbonate of lime, pyrites, mica, etc., etc., are found, then pieces of brick, such, for instance, as remained from the determination of soluble salts, are to be examined in a Papin's digester for their deleterious influence. They are to be so arranged

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* These sieves are the same as those used for the examination of cements (see below, VIII. *C* 2).

in a Papin's digester that they are not touched by the water directly, but are subjected to the action of the steam generated alone. The pressure of steam shall be one-fourth atmosphere and the duration of test, three hours. Possibly occurring disintegration is to be determined by means of the magnifying glass.

8. *For the determination of the relation between volumetric and superficial porosity, as also for the*

9. *Testing of roof tiles for their impermeability, the new Standing Committee is to develop propositions.*

C. Paving Stones and Ballast, Natural and Artificial.

1. Information in regard to petrographic and geologic *classification*, the *origin* of the samples, etc., etc.; also:

2. Statement in regard to *utilization* of same, precisely as under the same numbers of natural stones: B, α (see p. 54).

3. *Specific gravity* of the samples is to be determined.

4. All materials used in the construction of roads, provided they are not to be used under cover or in localities without frost, are to be tested for their *frost-resisting qualities*, by similar test to those prescribed for natural stone above (see *B*, α, No. 7–10).

5. *Flagging* or stone used for *sidewalks*, are tested most satisfactorily in a manner representing their mode of utilization, by determining the *wearing qualities* by a method as proposed by Professor Bauschinger and described in Part XI. of his "Mittheilungen." The uniformity of wearing qualities of burnt stone for parts more or less distant from the exterior surface is determined by repeating the trial on the same piece, and not merely testing *one*, but a *greater number* of pieces. It is, moreover, necessary to submit samples of the best, the poorest and the medium qualities of bricks in any one delivery.

6. For the determination of the value of stone for ballast (macadam) or paving purposes; the only decisive method is to construct trial roads, on which the materials employed are sure to receive a traffic as nearly uniform in kind and weight per metre of width as possible. It is urgently desirable that such trial pieces be numerously constructed by the supervisors of road construction in a uniform manner, and special attention is hereby called to the information given by Professor Dietrich in a paper entitled "Die Baumaterialien der Steinstrassen" (materials of construction of stone roads).

7. In order to determine the value of ballast and paving stone newly introduced in a manner more expeditious than that of the construction of trial roads, and so as not to increase the number of trial roads in proportion to the number of kinds of materials employed, it is necessary to determine a more rapid method of testing such stone. As ballast and paving materials are equally to be examined for resistance to abrasion and to frangibility, trials in a revolving drum, similar to those used in France for some time past, or others as described and delineated in the above mentioned pamphlet by Prof. Dietrich, are recommended. It appears, however, that an increase in diameter to 0.3 m., and of length to 0.5 m. is necessary, in order to increase the effect of impact. The rotary velocity as well might be increased; *the new Standing Committee is directed to investigate the most advisable rotary velocity, the quantity and dimensions of materials to be tested, and the method of determining the abrasion by actual practical trials, and to report results to the next conference.*

It is to be remarked, however, that the preparation of the trial ballast is not to be left to the client ordering the investigation, but is rather to be managed by the investigator, in order to secure greater uniformity.

It is a problem of experience to test the agreement of the results of this trial method, which ought to be preferred to the quarrying (boring) test (subject to continually varying conditions of the drill), with the results as obtained, by the trial roads themselves.

8. In addition to this drum test, ballast materials particularly are to be tested for their *resistance* to *compression*, as they are subject to crushing, and the blocks of stone for this purpose are to be cubes of from 5—7 c.m.

9. *Paving materials are to be examined for capacity of taking a polish (or becoming smooth), and the new Standing Committee is to submit propositions relative thereto to the next conference.*

10. In ballast and paving materials, also, it seems necessary to investigate the least, partly, and most suitable grades in a delivery of any one material, as uniformity of the structure made by them is one of the essential or primary qualities necessary.

11. Examinations of *asphalts* can only be made in an exhaustive manner by the construction of trial roads. An opinion coinciding with the results of such trial may be formed by

a. Determination of the quantity and quality of the bitumen contained therein (whether the bitumen be artificial or natural).

b. By physical and chemical determination of the residue.

c. By determination of the specific density of test pieces of the material used by a Vicat's needle of a circular sectional area of 1 sq. mm.

d. By the determination of the wear of such test pieces by abrasion or grinding trials.

e. By the determination of the resistance to frost of these test pieces.

D. Tests of Materials for Preservation of Natural and Artificial Building Stone.

1. The preservation of natural and artificial building stones shall be determined by the tension tests.

The results thus far obtained when determining the effect of materials for preservation, have uniformly shown that the resistance is increased, or at least that the tendency to deterioration under the effect of repeated freezing and thawing of saturated materials is diminished. As all materials for preservation take the form of coatings and do not permeate the stones to be examined, it appears that the strength is properly indicated by tension pieces of small section.

In these the ratio of surface to volume is greater, and the effect of the preservation is more easily determined.

2. For the form of the test pieces, the German standard of 5 c.m. minimum section is chosen.

All of the building stones requiring preservative treatment are soft, and can therefore be easily given the standard shape. For artificial stone the German mould can be used directly to produce the shape. The German tension apparatus is well adapted for the tests.

3. Three test pieces suffice for one series of tests. Should these, however, give decided variations, then an additional set of five pieces is to be tested.

4. The method prescribed for determining the resistance to frost as described under No. VII., *B*, α, and VII., *B*, β, for natural and artificial building stone, shall also be employed to determine the effect of preservative materials.

Also, it is recommended to determine the length or duration of the preservative effect. Practically this would be done by repeating the above tests after one and three and five years.

5. Inasmuch as the preservative effect may sometimes be due to the increased impermeability of the materials to the injurious components of the atmosphere, rather than an increased resistance, the

porosity or permeability of such would be shown by the capacity of absorbing water before and after treatment, and would be expressed in per cent. (%) of weight of water absorbed by a normal or standard test piece.

6. Test pieces to determine the preservative effect must be subjected to the exact treatment prescribed for actual practice. According to the nature of the preservative material the treatment must vary, as the manner of application may materially influence the effect.

VIII. TESTS OF HYDRAULIC MORTARS.

(Bond Materials.)

A. General Remarks.

1. When using hydraulic bond material for a specific purpose, such special use and also the mixing materials at hand (such as sand, gravel, slag, etc.) must always be borne in mind when making the tests, and used in the preparation of the specimen. These tests, however, are not to take the place of the so-called standard tests.

Sewer covers and pipes are to be tested by Prof. Bauschinger's method. (See Mittheilungen aus dem mech. techn. Laboratorium der techn. Hochschule in Muenchen, Heft. VII.)

2. The compressive and tensile resistance of hydraulic mortars, made in accordance with current standard methods, is not all sufficient to insure the durability of structures, as there are really several other important points to be considered as well; such as weathering qualities, brittleness, impermeability, adhesive strength, stability of volume of mortar, which are all of the greatest importance for the durability of structures.

Inasmuch as the strength of existing cement-mortars cannot be utilized, it appears superfluous from the point of view of the users of cement to still further increase it.

3. *The motion, " The admixtures of materials shall not in future be determined as heretofore by weight, but by volume," is submitted to the new Standing Committee.*

B. Nomenclature.

1. *Hydraulic limes* are products obtained by roasting (burning) limestones containing more or less clay (or silicic acid), and which

when moistened with water become wholly or partly pulverized and slaked. According to local circumstances, these are found in trade in lumps or in a hydrated condition in the shape of a fine flour.

2. *Roman cements* are products obtained by burning clayey lime marls below the temperature of decrepitation and which do not disintegrate upon being moistened, but must be powdered by mechanical means.

3. *Portland cements* are products obtained by burning clayey marls or artificial mixtures of materials containing clay and lime, at decrepitation temperature, and are then reduced to the fineness of flour, and which contain for one part of hydraulic material at least 1.7 parts of calcareous earth. To regulate properties technically important, an admixture of two per cent. of foreign matter is admissible.

4. *Hydraulic fluxes* are natural or artificial materials, which in general do not harden of themselves, but in presence of caustic lime, and then the same as a hydraulic material; *i. e.*, puzzuolana, santorine earth, trass produced from a proper kind of volcanic tufa, blast-furnace slag, burnt clay, etc., etc.

5. *Puzzuolana cements* are products obtained by most carefully mixing hydrates of lime pulverized, with hydraulic fluxes in the condition of dust.

6. *Mixed cements* are products obtained by most carefully mixing existing cements with proper fluxes. Such bond materials are to be particularly stated as "Mixed Cements," at the same time naming the base and the flux used.

C. Testing.

1. *Weight.*

a. The determination of the specific weight of hydraulic bond materials (specific weight of the granular parts) is to be determined uniformly by the apparatus called the "Volumenometer."

b. To determine the weight of a given volume of a hydraulic bond material, a cylindrical vessel 10 cm. high is to be used. Into this is to be

(α) sifted, and

(β) shaken,—the material to be weighed.

The particular method according to which each of these is to be done is committed to the new Standing Committee.

2. *Fineness of Grain.*

The fineness is to be determined by the use of sieves of 900 and 4,900 meshes per sq. cm. for Portland cements, and of 900 and 2,500 meshes (5,732 and 31,170 per sq. in.; 5,732 and 15,900 per sq. in.) per sq. cm. for the other hydraulic bond materials, and for each test a quantity of 1,000 gr. (2.22 lbs.) is to be used. The wires used in the above sieves shall have diameters of

0.05,	0.07,	and	0.1 mm. for sieves having
4,900,	2,500,	and	900 meshes per sq. cm.
(.00197,	.002758,		.00395 in.)
(5732,	15900,		31170 meshes per sq. in.);

and it is recommended to always procure the sieves from the same makers.

3. *Set.*

α. Of all hydraulic bond materials, excepting puzzuolana (trass).

a. Conditions of set are always to be investigated at a temperature of from 59°—65° F.

b. They are to be determined from a paste of *standard consistency*. To determine the latter the standard needle (see below) is used, combined with the "consistency meter," which is a rod of 1 cm. diameter, weighing 300 grammes (.66 lb.), and a cylindrical vessel 4 cm. high and 8 cm. diameter (1.576 × 3.152 in.) inside, made of a substance not permeable by water, and at the same time a bad heat conductor (preferably hard rubber).

To determine the standard consistency of a paste, 400 grs. of the material are stirred with a given quantity of water to make a stiff paste, which is worked by a spoon-shaped spatula for exactly three minutes for slow-setting and one minute for quick-setting materials; this is then filled into the vessel of the consistency meter without shaking. After stroking the surface of the paste the rod of the consistency meter is carefully inserted into the paste.

When this rod remains standing 6 mm. (.2364 in.) above the bottom of the vessel, the paste is said to have standard consistency.

c. The conditions of set are determined by the standard needle of 1 sq. mm. (0.000157 sq. in.) cylindrical section, and a weight of 300 grammes (0.66 lb.), and the same vessel as before.

Four hundred grammes (0.88 lb.) of the material are mixed with the quantity of water, determined as under *b*, and made into a paste, stirring slow-setting materials three minutes and quick-setting one minute, and with this paste fill the vessel flush. The material

has commenced to set when the needle no longer penetrates to the bottom. In quick-setting materials the commencement of set can also be determined by the thermometer.

To determine the *time of setting* the cake is removed from the vessel. Every hydraulic bond material can be said to have set when it has hardened so much that the standard needle no longer makes any impression. The time in which this degree of hardening occurs is called the *time of setting.*

Whether a material is to be classified as quick or slow setting is determined by the beginning of set.

d. As a *preliminary test* for the determination of the time of setting, the cake test can be employed. For this 100 grammes of the material to be tested are mixed with water to a paste of normal consistency, and stirred three minutes if slow, and one minute if quick setting; this paste is made into a paste of about 2 cm. thickness, and placed on a glass slab. The same is said to have set when it resists the finger nail under slight pressure.

e. Addendum.

It is desirable that the setting qualities be also determined with higher proportions of water than that necessary to make a paste of standard consistency.

β. For the determination of set of puzzuolano (trass) the following proposition is submitted to the new Standing Committee:

The finely powdered puzzuolano dried at from 212°—230° *F. is heated to redness to determine the loss by weight of combined water; the commencement of set is determined by means of the standard needle of* 1 *sq. mm.* (.00155 *sq. in.*) *section, weighing* 300 *grammes* (0.66 *lb.*) (*see above, α, c.*), *and when submerged in water of* 59° *F. as nearly as possible, but certainly with statement of temperature, using a mixture of two parts by weight of puzzuolano (trass), one part powdered hydrate of lime, and one part of water. The mortar thus obtained is to be filled into the standard vessel and immediately submerged, and examined after two, three, four, and five days to determine with what load the above standard needle enters to a depth of* 5 *mm.* (.197 *in.*).

4. *Constancy of Volume.*

α. Portland cement.

a. To obtain a rapid judgment of the constancy of volume of Portland cement when hardening under water or under conditions preventing drying, the simple baking test is to be resorted to, and as follows:

The cement is mixed with water to form a paste of standard consistency, and small cakes of from 8—10 cm. diameter and 2 cm. (3.94—3.15 in. diameter × .788 in. thick) in thickness are formed on flat glass plates.

Two of these cakes, which are guarded against complete drying to avoid shrinkage cracks, are placed after twenty-four hours, or at least after having set, with their smooth surfaces upon metal plates, and subjected to a temperature of from 230°—248° F. (at least one hour) until no more steam is given off.

If the cakes after this treatment show neither warping nor cracks on the edges, the material can be considered to have constancy of volume; otherwise the result of the now universal cake test on glass plates is to be made and the result obtained is to be decisive.

In the presence of more than 3% of dry sulphate of lime (or a corresponding proportion of unburnt gypsum) the baking test is not conclusive.

b. The deciding test of constancy of volume is the cake test on glass plates, which is to be executed as follows:

One hundred grammes (0.22 lb.) of the cement to be tested are mixed with water, and made into a paste of normal consistency and a cake thereof of about 2 cm. (0.788 in. thick) thickness is made on a plate of glass. Two of these cakes, guarded against drying, are submerged in water after twenty-four hours or when set, and the material is said to have constancy of volume when the cakes, after twenty-eight days' submersion, show neither warping nor cracking of edges.

β. Hydraulic Limes and Roman Cements.

For these the cake test, under water, as given under *α*, *b*, is recommended.

γ. Puzzuolano.

For these the following proposition is submitted to the new Standing Committee:

The determination of constancy of volume of puzzuolano mortar, consisting of two parts by weight of puzzuolano, one part powdered hydrate of lime, and one part by weight of water, is made by filling the vessel (see above, 3, *α*, *c.*) and stroking the same; then submerging it in water at least 2 cm. (0.788 in.) above the upper edge of this vessel.

The two halves of the vessel must not be spread by the material, nor must it rise above the upper edge.

Discovery of rapid methods for determining the constancy of volume of Portland cement, and other hydraulic bond materials in air, as well as the cooking test, are submitted to the new Standing Committee.

5. *Tests of Strength.*

α. For all hydraulic bond materials, excepting puzzuolano:

a. The tests of strength are to be made with mixtures of 1 part by weight of the material, and 3 parts by weight of sand. It is, however, desirable to make tests with a larger admixture of sand as well.

b. The sand to be used is to be standard sand, obtained from the purest quartz sand, the fineness of which is to be determined by using three sieves—64, 144, and 225 meshes per sq. cm. (407, 916, 1,431 per sq. in.)—so that one-half passes the first and remains on the second, and the other half passes the second and remains on the third sieve.

c. The wires of these sand sieves are to be as follows:

Sieves of	64	144	225 meshes per sq. cm.
Diam. of wire,	0.40	0.30	0.20 mm.
	407	916	1,431 meshes per sq. in.
	0.0157	0.01182	0.00788 in.

The Standing Committee is charged to determine whether sieves punched from plates can be substituted for wire sieves, and shall determine the size of holes to be used to correspond to above wire sieves.

d. The weight by volume of the standard sand is to be determined by weighing the sand sifted into a standard liter measure.

e. The several testing laboratories can use quartz sand of any source in order to obtain the standard sand; the prescribed fineness of grain must not, however, be varied.

When comparing results obtained by different laboratories, the standards are to be previously compared by mutual agreement.

The new Standing Committee is charged to determine a standard sand, not only in reference of grain, but also in regard to weight and all other influencing qualities.

f. The deciding test of resistance, determining its value, is the *compression test;* this is to be made with cubes of 50 sq. cm. (7.75 sq. in.) section.

g. The usual test of quality (test determining the quality of the material to be furnished) is the tension test, made by means of the

German standard apparatus, with test pieces of the German standard form, having a section of 5 sq. cm. (0.775 sq. in.) at the part to be ruptured.

h. The determination of a standard consistency of mortars, as well as the discovery of a practical mechanical device for producing test pieces, which, in particular, insures equal density of compression and tension test pieces, is delegated to the new Standing Committee.

For the present all test pieces are to be prepared by hand, having care to give them as nearly as possible equal density.

i. To carry out the tension and compression tests, six pieces of each age of mortar are necessary. The mean of the four highest results obtained is to be considered as the controlling one.

k. All test pieces must be kept in an air space saturated with watery vapor for the first twenty-four hours, after which, and until immediately before testing, submerged in water of a temperature 59°—65° F., to be changed once every seven days.

l. The critical test for all bond materials is to be the 28-day test.

To judge of the quality in *less time* in the case of Portland cement, the resistance may be determined upon mortar of the composition of 3 : 1 after setting 7 days.

Tests of resistance of pure Portland cements made up into a paste of normal consistency, on an impermeable plate, and those with standard sand after three days' setting, are delegated to the new Standing Committee.

The same is also to make propositions as to how other hydraulic bond materials may be examined rapidly for their quality.

β. All tests of resistance of puzzuolano mortars are to be made on mixtures by weight of 2 parts puzzuolano, 1 part powdered hydrate of lime, 3 parts standard sand, and 1 part of water. The treatment shall in other respects be the same as that of cement samples, and the puzzuolano mortars especially are to be kept in an air space saturated with watery vapor for 24 hours before being submerged.

For special tests the test pieces can be put under water immediately after making; in such cases it is recommended to increase the quantity of water by one-tenth when making the paste.

The observation of conditions of temperature is of greatest importance for puzzuolano mortars; if at all possible, the mixing and immersion water are to be kept at from 59°—65° F.

For test pieces of puzzuolano mortars only the purest air-slaked lime is to be used, as the strength depends largely upon the lime.

When builders manufacture their own puzzuolano from tufa then the stone is to be so finely powdered for sampling that 75 % will pass through the sieve of 900 meshes (5,732 per sq. in.) and 50 % through the sieve of 4,900 meshes per sq. cm. (31,170 per sq. in.), using wires of diameters stated above.

When powdering, no coarser particles are to be discarded, but the pulverization must be carried to such a point as to produce the prescribed fineness.

6. *Adhesion.*

The following propositions for the determination of adhesion are submitted to the new standing committee for examination and report.

The adhesion of all hydraulic bond materials and mortars is to be determined by the German standard testing machine, using test pieces enclosed in frames placed on dull ground-glass prisms. The test surface is to be $5 \times 5 = 25$ sq. cm. The tension holders must be entirely mobile; *i. e.*, a correct holding device for tension tests.

It is absolutely necessary that all test pieces of hydraulic bond materials be kept in a space saturated with water or submerged, as prescribed for tension tests.

In order to determine the adhesion of bond materials correctly, all accidental effects of the material upon the surfaces united must be excluded. Bricks, therefore, are never to be used when testing limes, and hydraulic limes and cements in particular, as they would form with lime a puzzuolano cement, which extends into the substance of the brick.

For practical reasons such substances are to be selected which can be treated with alkalies and acids without changing their composition. Really this leaves only ground glass and biscuit porcelain plates, and ground glass deserves preference for reasons of economy and as it is affected but very slightly by lime.

All ground-glass plates are to have the same finish.

7. *Density of Different Hydraulic Bond Materials for the Preparation of Mortars.*

The density is determined by the well-known mortar volumemeters or mathematically by Stahl's method (which is given in the XIVth. Part of the Mittheilungen aus dem mechan. techn. Laboratorium der kgl. techn. Hochschule, pp. 252 and 270).

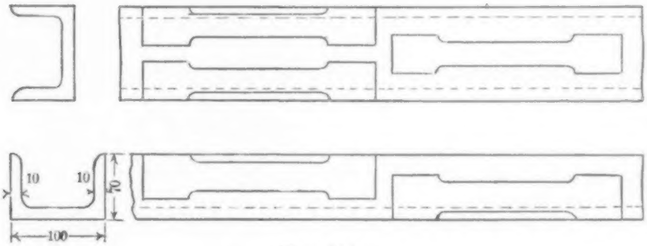

FIG. 106.

REGISTER.

Register of names of those gentlemen who took part in the deliberations of the conferences at Munich (M) and Dresden (D) and were members of the standing committee ($C_{I.}$ and $C_{II.}$), or who are still interested.

DELEGATES TO THE CONFERENCES.

Asimont, G., Prof. der kgl. techn. Hochschule München................ (M.)
Bach, C., Prof. der kgl. techn. Hochschule in Stuttgart u. Vorstand der Material-Prülungs-Anstalt das.................................... (M.)
Bauschinger, Joh., Prof. der kgl. techn. Hochschule in München und Vorstand des mech.-techn. Laboratoriums derselben.........(M, CI, D, CII.)
Belelubsky, N., Professor und Vorstand des mech.-technischen Laboratoriums am Wegebau-Ingenieur-Institut in St. Petersburg...(M, CI, D, CII.)
Bender, Rudolf, Ingenieur der öst.-ung. Staatseisenbahn-Gesellschaft in Wien.. (M.)
Berger, F., Stadtbaudirector in Wien ....(CI, D, CII.)
Bergk, kgl. Baurath an den sächs. Staatsbahnen in Chemnitz.......... (M.)
Bergmann, A., Cementf. in Linz, Oberösterreich...................... (M, CI.)
Berndt, R., kgl. Professor in Chemnitz............................ (M.)
Bernuolly, A., Cementfabr. in Wildau-Eberswalde .... (M.)
Blümcke, Dr., Assistent der Physik an der kgl. technischen Hochschule in München.. (CI.)
Böck, Rupert, k. k. Academie-Director in Leoben............... (M, CI, D, CII.)
Böcking, F. Oberingenieur des rhn. Dampfkessel-Ueberwachungs-Vereins in Düsseldorf.. (M.)
Böckmann, Baurath in Berlin .... (CI.)
Böhme, Dr., Vorstand d. k. Prüfungsanstalt für Baumaterialien in Berlin-Charlottenburg...(M, CI, D, CII.)
Bömches, Hafenbaudirector a. D. in Wien..........................(CI, D, CII.)
Brauer, Ernst, Prof. der tech. Hochsch. in Darmstadt................. (M.)
Brauns, Hüttendirector in Dortmund................................. (M, CI.)
Büssing, Ingenieur in Berlin S. W. 11, Bahnhofstr. 4 .... (M, CI.)
Bues, Karl, Architekt in Hamburg.................................... (CI.)
Coaz, Oberforst-Inspector in Bern.................................. (CI.)

Curti, Dr., Industr. in Winzendorf bei Wien.-Neustadt................ (M, CI.)
Dietrich, Professor der kgl. technischen Hochschule Berlin-Charlottenburg..(M, CI, D, CII.)
Dyckerhoff, Rud., Portland-Cementfabrikant in Amöneburg bei Biebrich a/Rh.. (M.)
Ebermayer, Gust., kgl. Rath bei der Generaldirektion der k. bayr. Staats-Eisenbahnen in München.................................. (CI.)
Eckermann, Oberingenieur des nordd. Dampfkessel-Ueberwachungs-Vereins in Hamburg.. (M, CI.)
Eisenbeis, Ingenieur in München.................................. (M.)
Erdmenger, Dr., Hannov. Portland-Cementfabrik Meisburg bei Hannover (CI.)
Erhardt, R., techn. Director der Neunkirchner Eisenwerke Gebr. Stumm in Neunkirchen bei Saarbrück.................................. (CII.)
Exner, Dr., Hofrath, Professor an der k. k. Hochschule für Bodencultur, Director des technolog. Gewerbemuseums in Wien............ (M. CI.)
Fischer, Hugo, Prof. des kgl. Polytechnik. in Dresden................ (D.)
Förster, Ritter v., Architekt in Lilienfeld............................. (CI.)
Frauenholz, W., Professor der kgl. technischen Hochschule in München (M.)
Gärtner, E., Ingenieur in Wien...................................... (D, CII.)
Gayer, Dr. Karl, Universitäts Professor in München.................. (CI.)
Gerber, H., Ingenieur, vo.m. Director der südd. Brückenbau-Actiengesellschaft in München.. (M, CI.)
Goedicke, Ed., Hütteningenieur, Donawitz bei Leoben jetzt Schwechat bei Wien...(M, CI, D, CII.)
Gollner, W., k. k. Professor der deutschen techn. Hochschule in Prag.(M, CI, CII.)
Gottgetreu, R., Prof. der kgl. techn. Hochsch. München.............. (M.)
Gottschaldt, Alw., kgl. Professor in Chemnitz......................... (M, D.)
Grau, Ad., kgl. Hütteningenieur in Regensburg......................... (M.)
Grass, Dr. O., Hüttendirector in Duisburg-Ruhrort..................... (M.)
Gyssling, W., Ingenieur und Director des bayr. Dampfkessel-Ueberwachungs-Vereins in München................................ (M, CI.)
Haage, A., Oberingenieur des sächs. Dampfkessel-Revisions-Vereins in Chemnitz.. (D.)
Haedicke, Director der Fachschulen und Lehrwerks-stätten in Remscheid.. (M, CI.)
Hartig, Dr. Regierungsrath, kgl. Professor am Polytechnikum in Dresden...(CI, D, CII.)
Haslinger, Th., techn. Director der Portland-Cementfabrik in Finkenwalde bei stettin.. (M.)
Hauenschild, Prof., Geologe u. Chemiker in Aarau...................... (M, CI.)
Heintzel, Dr. C., Laboratorium und Versuchsstation für die Cement-Industrie in Lüneburg.. (M, CI.)
Herfeldt, Gabr., Tuffsteingruben und Trassmühlen in Plaidt bei Andernach...(CI, D, CII.)
Hesse, Theodor, Fabrikant in Frankfurt.................................. (M.)
Hilpert, Ad., Director der Maschinenbau-Actiengesellschaft Nürnb., vorm. Klett und Co. in Nürnb.. (M, CI.)
Hoffmann, Regier.-Baumeister, Berlin N. Kesselstr. 7.................. (CI.)
Howaldt, Gg., Kieler Schiffswerft in Kiel..............................(CI, CII.)
Huber, H., Portl.-Cementfabrik in Rozloch, Schweiz..................... (M.)
Hübner, W., Maschinenfabrikant in Mannheim............................ (M.)

Jenny, K., k. k. Bergrath, Professor der technischen Hochschule, Wien ....(M, CI.)
Jenny, jun. Eisenbahningenieur in Wien ....(M.)
Kaemp, R. H., Maschinenfabrikant, Firma Nagel Kämp in Hamburg.... (D.)
Kayser, P., Architekt, Vorst. der Prüfungsanstalt für Baumaterial. an d. kgl. Baugewerbeschule in Dresden ....(D.)
Keim, Adolf, Chemiker und Redacteur in München ....(M, CI.)
Kessler, Franz, Ingenieur der k. k. priv. österr.-ung. Staats-Eisenbahn-Gesellschaft in Wien ....(M.)
Kerpely, A., Ritter v., Director der k. ung. Staats-eisenbahnwerke in Budapest ....(CI.)
Kick, Fr., k. k. Professor der deutschen techn. Hochschule in Prag..... (M, CI, D, CII.)
Klebe, C., Assistent des mech.-technischen Laboratoriums der kgl. techn. Hochschule in München ....(M, D.)
Krell, Metallfabrik in St. Petersburg, Wyborger Seite ....(CI.)
Lechner, C., Cementfabrikant in Oberkammerloh ....(M.)
Leuba, Arthur, Cementfabrikt. in Noiraigues (Schweiz) ....(CI.)
Leube, Dr. G., Cementfabrikant in Ulm ....(M, CI, CII.)
Leitzmann, L., Ingenieur in Radebeul bei Dresden ....(D.)
Lichtenfels, A. v., Betriebsdirector-Stellvertreter der österr. alpinen Montangesellschaft in Wien ....(CI.)
Lieven, Dr. V., Portlandcementfabrikant, Portkunda ....(D.)
Lismann, A., Kupferwerk in München ....(M, CI.)
Loewe, F., kgl. Prof. der techn. Hochsch. in München ....(M.)
Martens, A., Ingenieur, Vorstand der kgl. mech.-techn. Versuchanstalt in Berlin-Charlottenburg ....(M, CI, D, CII.)
Michaëlis, Dr. W., Cementtechniker in Berlin N. O. Friedensstr 15...... (M, CI, D, CII.)
Minssen, A., Ingenieur, Director in der Gussstahlfabrik Fr. Krupp in Essen ....(M, CI.)
Minssen, H., Oberingenieur des schles. Vereines zur Ueberwachung von Dampfkesseln in Breslau ....(M, CI, D, CII.)
Mohr, Mannh. Maschinenfabr. Mohr u. Federhaff Mannheim ....(CI.)
Moser, Ingenieur in Zürich (Riesbach) ....(CI.)
Nagy, D., Prof. der k. techn. Hochsch. in Budapest ....(M, CI.)
Nördlinger, Dr., kgl. Forstr., Univ.-Prof. in Tübingen ....(CI.)
Nonner, Hüttendirector in München ....(M, CI.)
Olschewsky, Hütteningenieur, Berlin N., Kesselstrasse 31...... (M, CI, D, CII.)
Osann, F., Ingenieur in Düsseldorf ....(M.)
Peters, Th., Ingenieur und Generalsecretär des Vereins deutscher Ingenieure, Berlin ....(M.)
Pfaff, Karl, k. k. Professor am technolog. Gewerbemuseum in Wien.... (M, CI, D, CII.)
Pittel, A. Baron v., Cementfabrikbesitzer in Wien ....(D.)
Pohlmeyer, Eisenbahndirector in Dortmund ....(M.)
Prásil, Fr., Ingenieur der Prager Eisenindustriegesellschaft in Kladno... (M.)
Pummer, Gust. A., Hütteningenieur in Neuberg ....(M, CI, D.)
Richter, Oscar, in Dresden ....(D.)
Rotter, Ed., Insp. der Kaiser-Ferd.-Nordb. in Wien ....(M, CI, D.)
Rziha, F. v., k. k. Prof. der techn. Hochschule in Wien ....(M, CI.)

Sailler, Alb., Oberingenieur in Witkowitz (Mähr) ....(M, CI, CII.)
Sattmann, Alex., Hütteningenieur in Prewali .... (M.)
Schifferdecker, Dr., Cementfabrikant in Heidelberg .... (M.)
Schön, Regierungsbaumeister in München .... (M.)
Schott, Dr. F., Fabrikdirector in Heidelberg ....(M, CI, CII.)
Schuchart, Hüttendirector in Wetter a/Ruhr ....(M, CI.)
Schulatschenko, Professor an der ingenieur-Academie in St. Petersburg. (CI, D.)
Schweizer, Director der Portland-Cementfabrik in Rozloch bei Luzern... (M.)
Seger, Dr. H., in Berlin-Charlottenburg .... (CI.)
Sickenberger, F., kgl. Oberbergrath a. D. in München .... (M.)
Skarbinsky, J. M., Ingenieur der Portland-Cementfabrik in Graduéc (Russ. Polen) .... (D.)
Smallenburg, F. W., Ingenieur in Zürich. .... (M.)
Stahl, B., Regierungsbaumeister in Frankfurt a/M ....(M, CI, CII.)
Stockert, L. Ritter v., Masch.-Ingen. in Wien.....................(M, CI, D.)
Stübben, Stadtbaumeister in Cöln a Rh .... (CI.)
Tetmajer, L., Professor u. Vorstand der eidgen. Anstalt zur Prüf. von Baumaterialien in Zürich ....(M, CI, D, CII.)
Toepfer, A. E., Portland-Cementfabrik Stern in Stettin .... (M.)
Tomei, Dr. A., Director d. Portland-Cementf. in Oppeln................ (M.)
Tröger, H., Romancementfabrik in Wallenstadt (Schweiz) .... (CI.)
Walther, Cementfabrik in St. Sulpiçe (Schweiz) .... (CI.)
Weizner, k. k. Oberingenieur, Pola.............................(CI, D, CII.)
Wigand, Paul, Verwalter der Vereinigten Cementwerke: Stuttgarter Cementfabrik Blaubeuren und Gebr. Leube Ulm, in Blaubeuren..... (M.)
Winkler, Dr. E., Professor der technischen Hochschule in Berlin........(CI, D.)
Wolff, E., Eisenbahnbaumeister a. D. in Berlin .... (D.)
Zervas, Wilh., in Firma Zervas D. Söhne, Tuffsteingruben, Trassmühlen etc. in Köln a/Rh .... (CI.)
Zwolenski, J., Oberingenieur der österreich.-ungar. Staatseisenbahngesellschaft, Domänendirection in Wien ....(M, CI, D.)

PAPERS

OF THE

CINCINNATI MEETING

(XXIst)

MAY, 1890.

CCCLXXIX.

PROCEEDINGS

OF THE

CINCINNATI MEETING

(XXIst)

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

May 13th to 16th, 1890.

LOCAL COMMITTEE:—Charles A. Bauer, J. R. Clay, Z. B. Coes, R. B. Collier, W. H. Doane, Joe. F. Firestone, Alexander Gordon, George A. Gray, *Chairman*, James C. Hobart, George Hornung, E. B. Johnston, Willis C. Jones, William H. Jones, John A. Keller, O. W. Kelly, Walter Laidlaw, H. M. Lane, O. A. Lanphear, Henry Marx, H. P. Minot, George W. Passel, S. W. Powel, A. W. Robinson, S. W. Robinson, James W. See, J. H. Springer, James B. Stanwood, *Secretary*, F. C. Trowbridge, J. H. Vaile, E. B. Wall, S. P. Watt, J. F. Webster, L. B. York.

FIRST DAY, TUESDAY, MAY 13TH.

The XXIst meeting of the American Society of Mechanical Engineers was called to order at 8 o'clock, in the Scottish Rite Cathedral of Cincinnati, Broadway, West Side, between Fourth and Fifth Streets.

Mr. Geo. A. Gray, as Chairman of the Local Committee, introduced Mr. M. E. Ingalls, President of the C. C. C. & I. R. R., who welcomed the Society. He was followed by Prof. H. T. Eddy, Dean of the University of Cincinnati. President Oberlin Smith, who was in the Chair, replied for the Society, after which the professional papers were taken up.

The first was that of Prof. J. E. Denton, entitled "Measurement of the Durability of Lubricants." This was discussed by Messrs. Almond, Porterfield, Hawkins, Scheffler, Dutton, Webb and Strong. The second paper was by Mr. C. S. Dutton, entitled "Some Experiences with Crane Chains," which was discussed

by G. C. Henning. The third paper of the evening was the "Memorandum" by Mr. Geo. H. Barrus, as to his taking indicator-cards from the engine of the steamer *City of Richmond*, which conveyed the party of American Engineers on their trip of May, 1889, from New York to Liverpool. At the close of this paper, the party adjourned to the Reception Rooms connected with the Assembly Hall, to enjoy an informal Reception with refreshments, tendered by the Local Committee.

Second Day, Wednesday, May 14th.

The second session was called to order at 10 o'clock in the morning in the Scottish Rite Cathedral. The Secretary's register showed the following members to be in attendance during the sessions of the Convention :

Ashworth, D .... Pittsburgh, Pa.
Almond, Thos. R .... Brooklyn, N. Y.
Ames, Prof. W. L .... Terre Haute, Ind.
Beck, M. A .... East Saginaw, Mich.
Ball, F. H .... Erie, Pa.
Barnard, Geo. A .... New York City.
Barrus, Geo. H .... Boston, Mass.
Binsse, Henry L .... New York City.
Borden, Thos. J. (*Vice-President*) .... Fall River, Mass.
Brown, Alex. T .... Syracuse, N. Y.
Barnaby, Chas. W .... St. Louis, Mo.
Bond, Geo. M. (*Manager*) .... Hartford, Conn.
Burgdorff, T. F. (*U. S. N.*) .... Knoxville, Tenn.
Bauer, Chas. A .... Springfield, O.
Baugh, S. A .... Detroit, Mich.
Brown, C. S .... Terre Haute, Ind.
Barnes, David L .... Chicago, Ill.
Barnes, W. F .... Rockford, Ill.
Blair, H. P .... Rochester, N. Y.
Bulkley, Henry W .... New York City
Baldwin, Bert. L .... Cincinnati, O.
Cooper, Henry R .... Syracuse, N. Y.
Cullen, Jas. K .... Chicago, Ill.
Colwell, A. W .... New York City.
Clay, John R .... Dayton, O.
Carpenter, Prof. R. C .... Lansing, Mich.
Cole, J. Wendell .... Columbus, O.
Crocker, John B .... Hamilton, O.
Crouthers, J. A .... New York City.
Cooley, Prof. M. E .... Ann Arbor, Mich.
Dock, Herman .... Philadelphia, Pa.

Dashiell, Benj. J., Jr	Baltimore, Md.
Doran, W. S	New York City.
Durfee, W. F	Birdsboro, Pa.
Derbyshire, W. H	Philadelphia, Pa.
Dick, John	Meadville, Pa.
Dutton, C. Seymour	Youngstown, O.
Downe, H. S	Chicago, Ill.
Dodds, E	Indianapolis, Ind.
Darlington, F. G	Indianapolis, Ind.
Eberhardt, F. L'E	Newark, N. J.
Forsyth, Wm. (*Manager*)	Aurora, Ill.
Forney, M. N	New York City.
Fawcett, E	Alliance, O.
Firestone, Joe. F	Columbus, O.
Gould, W. V	Norwich, Conn.
Greenwood, J. H	Cleveland, O.
Gobeille, Joseph Leon	Cleveland, O.
Gilmore, Robt. J	Providence, R. I.
Gill, John L	Philadelphia, Pa.
Geer, J. H	Johnstown, Pa.
Gray, Prof. Thos	Terre Haute, Ind.
Gale, Prof. Horace B	St. Louis, Mo.
Gray, G. A	Cincinnati, O.
Haskins, Harry S	Philadelphia, Pa.
Heggem, Chas. O	Massillon, O.
Holmes, Isaac V	Beloit, Wis.
Henning, Gus. C	New York City.
Hibbard, H. D	Pittsburgh, Pa.
Hill, Wm	Collinsville, Conn.
Hunt, C. W	New York City.
Hobart, Jas. C	Cincinnati, O.
Hawkins, John T	Taunton, Mass.
Henthorn, John T	Providence, R. I.
Higgins, S	Meadville, Pa.
Hunter, Geo. E	Elgin, Ill.
Herman, L	Cleveland, O.
Hornung, Geo	Cincinnati, O.
Ide, A. L	Springfield, O.
Jones, W. H	Cincinnati, O.
Jones, W. C	Cincinnati, O.
Jacobus, Prof. D. S	Hoboken, N. J.
Jenks, W. H	Brookville.
Johnston, Ed. B	Cincinnati, O.
Kempsmith, F	Milwaukee, Wis.
Kirk, W. A. L	Chattanooga, Tenn.
Kirby, Frank E	Detroit, Mich.
Kinder, J. J. De	Philadelphia, Pa.
Kirkevaag, Peter	Youngstown, O.
King, C. I	Madison, Wis.
McKinney, R. C	Hamilton, O.
Kelly, O. W	Springfield, O.

Keller, J. A	Hamilton, O.
Lanphear, O. A	Cincinnati, O.
Laird, John A	St. Louis, Mo.
Lipe, C. E	Syracuse, N. Y.
Lane, H. M	Cincinnati, O.
Laidlaw, Walter	Cincinnati, O.
Landreth, Prof. Olin H	Nashville, Tenn.
Morris, Henry G	Philadelphia, Pa.
Magruder, Prof. Wm. T	Nashville, Tenn.
Mack, J. G. D	Cincinnati, O.
Miller, Jas. S	Erie, Pa.
Morgan, T. R., Sr	Alliance, O.
Moore, A. G	Cincinnati, O.
Moore, E. L	Portsmouth, N. H.
McFarren, S. J	McKeesport, Pa.
Marx, Henry	Cincinnati, O.
March, P. G	Cincinnati, O.
Miller, Fred. J	New York City.
Nagle, A. F	Chicago, Ill.
Nason, Carleton W. (*Manager*)	New York City.
Passel, Geo. W	Cincinnati, O.
Percival, Geo. S	New York City.
Penney, Edgar	Waynesboro, Pa.
Porterfield, H. O	Pittsburgh, Pa.
Parks, E. H	Providence, R. I.
Powel, S. W	Hamilton, O.
Penruddock, J. H	Fort Gratiot, Mich.
Pickering, Thos. R	Connecticut.
Rogers, W. S	Troy, N. Y.
Robb, D. W	Amherst, Nova Scotia.
Raynal, A. H	Richmond, Va.
Roberts, T. H	Detroit, Mich.
Roberts, E. P	Cleveland, O.
Robinson, J. M	New York City.
Robinson, A. W	Bucyrus, O.
Robinson, Prof. S. W	Columbus, O.
Randolph, L. S	Baltimore, Md.
Smith, Oberlin (*President*)	Bridgeton, N. J.
Stanwood, J. B	Cincinnati, O.
Suplee, H. H	Philadelphia, Pa.
Spies, Albert	New York City.
Smith, Jesse M	Detroit, Mich.
Smith, Geo. H	Providence, R. I.
Spangler, Prof. H. W. (*U. S. N.*)	Philadelphia, Pa.
Smith, Chas. P	Norwich, Conn.
Stillman, F. H	New York City.
Spaulding, H. C	Boston, Mass.
Smith, C. M. W	Dunkirk, N. Y.
Stiles, N. C	Middletown, Conn.
See, J. W	Hamilton, O.
Sancton, E. K	Scranton, Pa.

Springer, Jos. H	Hamilton, O.
Sowter, J. G	Detroit, Mich.
Smith, Chas. F	Newark, N. J.
Scheffler, Fred'k A	Pittsburgh, Pa.
Samuels, J. H	Moline, Ill.
Sweeney, John M	Wheeling, W. Va.
Swasey, Ambrose	Cleveland, O.
Thomson, John	New York City.
Tilden, James A	Boston, Mass.
Underwood, F. H	Connecticut.
Vaile, J. H	Dayton, O.
Woodbury, C. J. H. (*Acting-Secretary*)	Boston, Mass.
Webb, Prof. J. B	Hoboken, N. J.
Whitlock, Prof. R. H	Brazos Co., Texas.
Wood, Walter	Philadelphia, Pa.
Webster, J. H	Boston, Mass.
Warrington, Jesse	Indianapolis, Ind.
Watt, S. P	Columbus, O.
Whitney, Baxter D	Winchendon, Mass.
Whitney, Wm. M	Winchendon, Mass.
Weeks, Geo. W. (*Manager*)	Clinton, Mass.
Webster, J. F	Springfield, O.
Warner, W. R	Cleveland, O.
Woods, Prof. A. T	Champaign, Ill.
Warren, B. H	Boston, Mass.
Wiley, Wm. H. (*Treasurer*)	New York City.
Webber, Wm. O	Erie, Pa.
Wightman, D. A	Pittsburgh, Pa.
Watts, Geo. W	Philadelphia, Pa.
Wall, Ed. B	Columbus, O.
Wright, L. S	Philadelphia, Pa.
York, L. D	Portsmouth, O.

The first business was the report of the Council, which was submitted as follows:

REPORT OF THE COUNCIL.

The Council would present its semi-annual report to the Society as follows:

The loss by death since the report at the New York meeting in November has been:

Mr. Fred. B. Rice.
Mr. Hector C. Havermeyer.
Mr. C. A. Ashburner.
Mr. Chas. D. Smith.
Mr. Horatio Allen, Honorary Member.
Mr. Gustav Adolph Hirn, Honorary Member.

The present membership, including the result of the election reported herewith, is distributed as follows:

Honorary Members	16
Life Members	9
Members	956
Associates	48
Juniors	104
Total	1,133

The Council also congratulates the members upon the consummation of its plans to secure for the Society's use permanent and adequate accommodations in New York City. The Council has made arrangements by which the Society will enter upon the occupancy of part of the house No. 12 West Thirty-first Street, New York City, which has been purchased and is held by trustees under an arrangement which gives the Society a permanent location at this place. The house is exceptionally large for a New York City house, being twenty-eight feet four inches wide, and with the auditorium or assembly-hall is ninety-eight feet deep.

The location is unexcelled, being between Fifth Ave. and Broadway, close to the group of uptown hotels and within two blocks of the new Engineers Club, which is directly in its rear. It stands two hundred feet west of Fifth Ave., with stations on the elevated roads, on the Sixth Ave. line, at Twenty-eighth and Thirty-third Streets, and on the Third Ave. line, at Twenty-eighth and Thirty-fourth Streets. Stage and car routes are also on Fifth Ave. and Broadway.

The auditorium is two stories high and has a seating capacity for over two hundred, and will be lighted with electric incandescence lamps, and provision for adequate mechanical ventilation will be made. The house will be newly decorated before the Society begins its occupancy, and many additional facilities will be furnished.

The business offices of the Society will be the large parlor of the house, at the right of the entrance. Its library will be in the large saloon on the second floor, and in the gallery of the auditorium, which surrounds it on the second floor.

The third story will contain an additional room for the use of members, where a drawing-table and other special facilities of

this kind will be furnished for the use of non-resident members when in New York.

For the present, in order to meet the financial burden imposed during the first years of occupancy of the house, the top floor and the basement will be leased to other parties. The American Institute of Electrical Engineers will also come into the building as tenants—their members with ours having the privileges of the growing engineering library. The library opening in the evenings, which was inaugurated at the New York meeting, has been very successful and popular during the winter, and will be continued until further notice. Access may be had to it between the hours of 10 A.M. and 10 P.M., every day except Sundays and the usual legal holidays. The interest of members of the Society is again solicited to secure a more rapid development of the library feature of the Society's life, particularly in the way of contributions of books, pamphlets, manuscripts, photographs, and other similar material.

The cost of the house into which the Society has arranged to enter was $60,000, of which $27,000 has been loaned to the Trustees by the membership, by subscription to bonds issued by that body, bearing interest at 5 per cent. The generous response of the membership to the request for money to inaugurate this new feature of the Society's growth and development was most gratifying. Over $10,000 was subscribed beyond what was necessary to be paid in cash to secure the transfer of the property. A "Sinking Fund" has also been created by the members, by which these bonds will be bought in and become the property of the Corporation, instead of being owned by individual members as at the start. The contributions to this Sinking Fund range from $2 to $25, many of the bond-holders contributing the amount of their interest for this Redemption Fund. It is urged that as many as possible who are not now contributors to this Sinking Fund, should hasten to become so in order to diminish the interest account as rapidly as possible, and to expedite the transfer to the Corporation of the unincumbered title to the property.

The Council has held more meetings than usual this past winter, caused not only by reason of the routine business and the acting upon an unusual number of applications, but the question of permanent quarters has been in agitation since the very first of the year. Much of the result of its action has been

made public in circulars which have been sent to the membership from time to time, or in the Reports of the Committees to be presented at this meeting.

The Council would further present the Report of its Tellers of Election as follows :

REPORT OF THE TELLERS, XXIST MEETING.

The Council herewith presents the report of its Tellers as folllows :

The undersigned were appointed a Committee of the Council to act as Tellers, under Rule 13, to count and scrutinize the ballots cast for and against the candidates proposed for membership in the Society of Mechanical Engineers, and seeking election before the XXIst meeting of the Society, in May, 1890.

They would report that they have met upon the designated days in the office of the Secretary, and proceeded to the discharge of their duties.

They would certify for formal insertion in the records of the Society, to the election of the appended named persons, to their respective grades, upon Lists Nos. 1 and 2, respectively pink and yellow.

There were 417 votes cast in the ballot upon the pink list, of which 14 were thrown out because of informalities.

There were 404 cast upon the yellow ballot, of which 13 were thrown out because of informalities. The lists are appended below.

Carleton W. Nason, } *Tellers.*
Wm. H. Wiley, }

FOR HONORARY MEMBERSHIP.

Porter, Chas. T.

AS MEMBERS.

Agassiz, A.	Clarke, Thos. Curtis.	Davis, Chester B.
Baldwin, Bert. L.	Clark, Walton.	Delaney, Alexander.
Boies, H. M.	Clawson, Linus P.	Derbyshire, Wm. H.
Bristol, Wm. H.	Coleman, John A.	Drysdale, Wm. A.
Brown, Alex. T.	Corry, Wm.	Ehbets, C. J.
Brown, Chas. S.	Crocker, John B.	Ehlers, Peter.
Canning, Wm. P.	Darlington, Frank G.	Gibbs, Geo.
Carter, Vaulx.	Dashiell, Wm. W.	Gray, Thos.
Cité, Joseph D.	Davis, Chas. H.	Graves, Erwin.

Green, Samuel W.	Martinez, Manuel J.	Richter, Ernst.
Gregory, Wm.	Mellen, Edwin D.	Riesenberger, Adam.
Grimm, Paul H.	Mellin, Carl J.	Sharples, Philip M.
Harris, John H.	Miller, Fred. J.	Sheldon, Frank P.
Hilles, T. Allen.	Moore, A. G.	Shirrell, David.
Hunter, Geo. E.	Moore, Douglas C.	Smith, Thos. G., Jr.
Jarvis, Samuel E.	Muller, Edward A.	Smith, Walter W.
Johnsen, Carl A.	Neave, Joseph S.	Snow, Sylvester M.
Johnson, Nils.	Nichols, Frank L.	Sornborger, Edwin C.
Jones, Edwin H.	Otis, Spencer.	Spilsbury, E. G.
Jones, Forrest R.	Peck, Staunton B.	Stockham, Wm H.
Jones, John T.	Perry, Nelson W.	Torrey, Herbert Gray.
Kneass, Strickland L.	Pouten, Anders.	Towl, Forrest M.
Lape, Wm. E.	Porterfield, H'y A.	Tynan, J. W.
Leonard, Arthur.	Price, John A.	Van Atta, Harry.
Lewis, Rollin C. W.	Ramsay, Jas. D.	Wood E. J.
Lodge, Wm.	Reiss, Geo. T.	Wood, Matt. P.
Mack, John G. D.	Rice, Alva C.	Wright, Louis S.
McKinney, Robt. C.	Rice, Richard H.	Zehnder, Chas. H.
Margedant, Wm. C.		

FOR PROMOTION TO FULL MEMBERSHIP.

Caldwell, A. J.	Sinclair, Geo. M.	Trump, Edward N.
Cruikshank, Barton.	Tompkins, S.	Wood, Walter.
Platt. John.		

AS ASSOCIATES.

March, P. G.	Pomeroy, Lewis R.	Redwood, Iltyd I.

AS JUNIORS.

Anderson, Larz. W.	Dashiell, Benj. J., Jr.	Ross, Edward G.
Bates, Albert H.	Dawes, Robt.	Thomas, Edward G
Bullock, Edwin R.	Dinkel, Geo.	Waldron, Fred. A.
Churchill, W. W.	Eilers, Karl.	
Cooper, Henry R.	Magoun, H'y A.	

The report of a Special Committee appointed by the Council to memorialize the Congress of the United States in the matter of a suitable recognition by the Nation of the work of Capt. John Ericsson, late member of this Society, reported through its Chairman the following letter and draft of such memorial :

ITHACA, N. Y., *May 1st*, 1890.

TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS :

GENTLEMEN :

I have the honor to submit, herewith, a memorial prepared by the Committee appointed by you to memorialize Congress relative to the erection, by the General Government, of a monument in memory of our recently deceased colleague, the great engineer and inventor, John Ericsson.

Your committee have prepared this memorial, and desire to lay it before the Society as a report of progress, with the suggestion that it be printed and presented to Congress in such a manner as may be determined to be most likely to accomplish the object for which your Committee is appointed. Authority being conferred upon the Committee to proceed, steps will at once be taken to lay the matter before Congress.

Very respectfully,

R. H. THURSTON,
Chairman.

For the Committee.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,
NEW YORK CITY, 1890.

TO THE HONORABLE THE SENATE AND HOUSE OF REPRESENTATIVES OF THE UNITED STATES OF AMERICA, IN CONGRESS ASSEMBLED.

The undersigned, a Committee of the American Society of Mechanical Engineers, appointed to represent that body as memorialists, respectfully submit the following statement and request :

In the death of John Ericsson, one of the greatest inventors and mechanicians of the last generation and of the present century, as well a member of this Society, and one of our most distinguished colleagues, our country has lost one of its greatest benefactors, the profession one of its noblest members, and the world one whose genius has seldom been equalled and has never been surpassed.

When the integrity of this, his beloved adopted country, was threatened by internal dissensions, this already famous engineer dropped every other project and gave his whole time and every power of his mind and body to the production of an engine of war that was to effect a complete revolution in modern methods of naval construction and which, as all know, appeared upon the scene of action at an instant when everything seemed lost, beating back the foe and saving the remnant of our fleet, thereby preserving very possibly the Union itself. This extraordinary vessel, called by Ericsson the *Monitor*, with the expressed intention of admonishing not alone this country, but the whole world that methods of naval warfare would be changed, was designed complete in an extraordinarily short time. The hull was built by Thomas F. Rowland of the Continental Works and the engines and many important details constructed by Ericsson's life-long friend and financial supporter, the late Cornelius H. Delamater. The vessel was built and launched in one hundred days, with steam already raised. She was sent to sea in an emergency almost without trial, and next day steamed into Hampton Roads, where the *Merrimac* had already showed herself invincible in a contest with our wooden fleet and when it was believed by those who witnessed her fearful attack upon the *Congress* and the *Cumberland* that no human power could avail to prevent the complete destruction of our fleets, and the raising of the blockade upon the southern sea-ports. The results of the introduction of this new and strange vessel upon the sea need not be detailed. The repulse of the *Merrimac* by the *Monitor* was an essential factor of our success in repressing the rebellion. What might have been the consequences if she had not appeared at that time have been a thousand times described by military critics. Your memorialists simply desire to ask that the work of that day and its far-reaching results be remembered, and that the nation, by its re-

presentatives in Congress assembled, show its gratitude to the great inventor and patriot, who did more for the country to preserve its fleets, protect its armies, preserve the blockade and to save the nation from disruption than all its forces combined could have done without the aid of Ericsson's genius. But a day before, apparently the resources of the world were impotent to avert the threatened blow; but the head, the heart and the hand of one great patriot and engineer proved equal to the accomplishment of the apparently impossible. Our fleets were saved, our armies continued to receive the support of the navy, the blockade was maintained upon our coasts and the possibility of preserving this nation from disruption was made a certainty. From that memorable day on which the *Monitor* met the *Merrimac* and compelled her destruction, no serious fear was again entertained of the breaking of the blockade, of the failure of the support of the army by the navy, or of the more dreaded interference of any foreign power in favor of the States in rebellion. That day the United States of America reasserted her position before the world, as on the one hand a peaceful nation, threatening no other, but invincible on her own shores, even in the face of the most serious internal dissensions, and fearing no combination of the most powerful of foreign nations. We submit that John Ericsson did more than any other one man or body of men did or could to give this nation absolute independence, absolute safety and wholesome respect in the eyes of the civilized world.

The single achievement above referred to should be sufficient to warrant this body in asking any possible token of respect and esteem from a grateful nation. We would, however, briefly refer to other works of this most extraordinary engineer.

Familiar with every aspect of steam engineering, his remarkable locomotive, the "Novelty," superior in some respects to even the historic "Rocket" of George Stephenson, illustrated just sixty years ago his wonderful fertility of invention as well as his intimate knowledge of the principles involved in the application of steam for various purposes.

Coming to this country a half century ago at the solicitation of Com. Stockton, one of the most far-seeing naval officers of the time, he brought to us the invention which has revolutionized modern naval construction, the screw propeller; and the U.S.S. *Princeton* was the first of a fleet that became and remained until the outbreak of our civil war the best navy afloat. In giving us the screw propeller, the great inventor conferred upon us an advantage which it is impossible to overestimate; but he was himself never paid for his office labor, and the government which he later did so much to save has never yet relieved itself even of the money obligation then incurred.

His complete experiments with the "hot air" or so-called "caloric" engine showed to the world the possibilities and limitations of that system, and resulted in providing a source of power which can be safely utilized in any location by those unfamiliar with the technical details of management of steam-engines and other motors, and has formed the basis of an important and far-reaching industry. His more recent work, designed to utilize directly the heat of the sun by means of his sun motor, showed his love of investigation to ascertain scientific truths rather than for his own advantage, and there is no doubt that the labor expended in this direction, illustrating some of the highest attainments of his great genius, will be utilized under modified conditions in the development of great inventions in the future.

His foresight in the construction of the *Monitor*, which would be invulnerable to ordinary attacks above the surface of the water, led him to provide for the

future, when the wealthier foreign nations should have constructed enormous vessels embodying the principles of the *Monitor*, but of sufficient power to threaten our own with destruction. To accomplish this, he developed and constructed some years since the *Destroyer*,* another style of vessel, capable of destroying a *Monitor* or any other form of iron-clad by attacking it with submarine projectiles, which would reach vital parts below water where protection was impossible. The *Destroyer* has not received the attention and encouragement its merits demanded, simply because the former invention of the *Monitor* has made the days of peace so lasting that there has been no necessity of utilizing his next great step in the development of modern naval warfare.

We are instructed by the American Society of Mechanical Engineers to request that Congress recognize the eminent and inestimable services rendered our country by this great engineer, and take action in such manner as may seem best and most appropriate to commemorate the work of John Ericsson and the part which he took in the preservation of our national union, and to show the appreciation of the nation of his genius and industry in the development of his great inventions and discoveries, which made their impress upon the time in which he lived and have already proved a blessing to all mankind.

In the province of Wormland, Sweden, there stands a monument built by the people of his native village ; a shaft of granite, on which is inscribed the legend : "John Ericsson was born here in 1803." Foreign monarchs and governments have conferred upon our fellow-citizen and compatriot honors and orders of every degree and in great numbers ; every great nation has vied with every other to do him honor ; he was elected to honorary membership in the great scientific and engineering societies of Europe and America ; he was surfeited with that attention which comes of great fame. Humble workmen, great nations, private citizens, powerful government officials, at home and abroad, recognized his genius, admired his works, appreciated his patriotism and those greatest achievements inspired by it ; but only his adopted country and his own government have failed to exhibit in any manner, even in the most insignificant way, that gratitude and that respect which he has so well earned and which should have been his in such immeasurable degree. John Ericsson died unhonored where he should have received most reverence, and his own country, which he did so much to save, is still in his debt. It is this wrong that we are appointed to ask the government of his country to right ; it is that honor, shamefully late though it may be, that we are desired to ask for our comrade of the greatest of his innumerable beneficiaries, our own country.

All of which is respectfully submitted for the American Society of Mechanical Engineers.

R. H. THURSTON,
CHAS. E. EMERY,
J. F. HOLLOWAY, } *Committee.*
GEO. H. ROBINSON,
E. D. LEAVITT,

On motion of Mr. B. H. Warren, the draft of the Committee was accepted and adopted as the sentiment of the Society.

The printed reports of the Committees on Uniform Methods of Conducting Duty Trials of Pumping Engines, and of the Committee on Uniform Methods of Test, were presented and

accepted. In presenting the report of the Committee on Uniform Methods of Test, the following discussion was had:

Mr. G. C. Henning.—I should like to say a few words about this report. The report unfortunately was not ready for distribution before the meeting. The appendix was received a few days ago. The report is presented provisionally for the purpose of obtaining a written discussion by all those who are interested in this matter. The Committee proposes to send copies of the report to members of the Society and non-members, to engineers and specialists interested, abroad as well as in this country, to obtain their views; and then to continue their work, if desirable, in order to propose standards universally acceptable, if possible. The only definite recommendations that can be made now which have been mentioned in the paper are those which refer to standard or scientific tests. The report states this, but if it is not read very carefully this point might be overlooked; and, therefore, I wish to call particular attention to it. The Committee has not yet been able to make general recommendations for routine or shop tests, such as are carried on almost everywhere in this country in shops and mills; the recommendations made and shapes of test pieces given in this report are at present intended solely for standard or scientific testing. There may be other standards advisable for routine testing, because neither the time nor money can be spent in routine testing which must be spent on scientific tests, or the preparation of test pieces; in such there must be no differences as to shape or size, or character, in any way. In scientific or standard testing too much pains cannot be taken in preparing the preliminary matters, of whatever kind they are. In routine testing these matters are frequently overlooked, and the object of the Committee is now to collate as much material on the subject of routine testing as will show whether a general method can be formulated. The Committee has also found, upon interviews with the prominent specialists engaged in testing, in Austria, Germany, France, Belgium, Switzerland and Great Britain, that the methods adopted for routine testing in this country are almost identical with those that have been in use abroad. Consequently there seems to be comparatively little trouble to propose standards which may be international. But the Committee has not had time, as I have before stated, to settle definitely upon this, or to embody it in this report, and for that

reason the members of the Society are particularly requested to look into this report and send in their discussion in writing. There is so much matter contained in a report of this kind that if each paragraph were taken up it would probably call forth lengthy disscussion. So it seems better to present the report provisionally that those who are interested in the work can study it at leisure, and then discuss it intelligently and at length.

On motion, the report was accepted and the Committee continued.

Mr. Wm. Forsyth.—The subject of this report is standard tests and method of testing. That is a very general term. If the work of this committee is to be confined to methods of testing materials, it seems to me that the word "materials" should be added to the title in order to distinguish it from the work of Committees, on testing other things. I would, therefore, move that the word "materials" be added.

The President.—Then, your motion is that the name of this Committee be altered to read, "Report of Committee on Standard Test and Methods of Testing Materials?"

Mr. Henning.—I would say that this Committee has always been known as indicated by the title. I think myself it would be advisable to say that the Committee is a Committee on Standard Tests and Methods of Testing Materials, because that includes not only materials of construction, as generally understood, but woods, alloys, and the soft metals. To add the word "materials" would be sufficient to characterize the Committee work fully.

The President.—It is moved that the word "materials" be appended to the name of this Committee.

The motion was seconded, and duly carried.

Later in the meeting, a discussion by Mr. A. M. Wellington, on the report on the method of conducting duty trials of pumping engines, was presented, and Mr. Geo. H. Barrus, its Chairman, presented the following request:

Mr. Geo. H. Barrus.—As Chairman of the Committee on Duty Trials, I would like to say that I am very glad to have this discussion, and I am sure that the other members of the Committee will agree with me. I wish there could be a more general discussion of the subject before it is left. It is not impossible that there are other matters to which some have taken exception, and

which might be brought out at this meeting, and I would be very glad personally, and I think the other members of the Committee would be, too, to hear what the members have to say upon it.

In view of these considerations, it was suggested that both Committees continue, and that the members be solicited to present further suggestions and comments. The reports as presented are printed as papers of the meeting, but they will be again presented at a future meeting for final acceptance as the work of the Committees, with such discussions appended to them.

The report of progress of the Society's Committee on Standard Flange Diameters of Pumps, they then presented as follows:

PHILADELPHIA, PA., *April* 6th, 1890.

TO THE PRESIDENT AND MEMBERS AMERICAN SOCIETY MECHANICAL ENGINEERS.

GENTLEMEN:

The Committee on Flange Diameters of Pumps, Valves and Pipes, begs to submit the following report: Owing to the pressure of other matters requiring the attention of your Chairman, and changes in the addresses of some of the members, which interrupted communication with them, no active steps were taken until Dec., 1889, when a circular (a copy of which is appended) was sent to forty-eight manufacturers and agents of pumps, valves, etc. The circular contained three leading questions. The first asked if the party addressed had adopted a standard, and, if so, would they send a copy. The second asked what special reasons governed the selection of their standard. Third, if they would be willing to modify their standard, to conform to one that might be recommended by the Society. Replies to the circular have been received from eleven firms, brief extracts from which are here given:

No. 1. "Would modify standard if large manufacturers would agree to do the same."

No. 2. "Is gratified to learn of the movement and wishes success."

No. 3. "Could not change, except by united combination of a considerable part of their consumers."

No. 4. "Have adopted Morris & Taskers' list; could not have standards on account of varying pressures of pumps."

No. 5. "Would welcome any standard the Society would agree to."

No. 6. "Thinks both classes of valves (gas and water) should have the same size of flanges, etc."

No. 7. "Asks for all available information concerning standards."

No. 8. "Thinks uniformity would be a decided benefit, but that the cost would not warrant the attempt."

No. 9. "Are glad to give attention to the matter, but would prefer not to make any change."

No. 10. "Have no special standard, but if the Society would hit upon one, would be pleased to coöperate. Have felt the necessity of a standard."

No. 11. "Wishes that a standard could be adopted, and would be willing to change theirs."

Five of the eleven firms enclosed copies of their standards, and the letters received are appended.

From the small percentage of replies received, the Committee does not think it advisable to compile a standard for presentation to the Society at this time, and asks to be continued. In view of the delay in getting to work, we feel that a formal report of progress would not be as satisfactory as a statement of what has been done. No meeting of the Committee has been held, owing partly to the distances apart of the members, and also to the lack of business important enough to bring them together. At the same time, we feel that the subject is important enough not to be hastily abandoned, and it is our intention, if continued, to supplement the circular already sent out, with another, making a strong appeal for co-operation. If successful, we hope to be able to present a final report at the next meeting of the Society.

Respectfully,

PERCY A. SANGUINETTI, *Chairman.*
A. H. RAYNAL,
SAMUEL S. WEBBER.

Copy of circular appended :

PHILADELPHIA, *Dec.* 20th, 1889.

DEAR SIR :

At a recent meeting of the American Society of Mechanical Engineers, I read a paper on the divergencies in the flange diameters of pumps, valves, pipes, etc., in which I brought to the notice of the Society the inconveniences and loss of time to the users of these various products, owing to the want of uniformity in the standard adopted by different makers.

The paper was discussed and a Committee appointed to confer with the manufacturers to ascertain their views and preferences, and report, if possible, upon a uniform standard, the enquiry to embrace also the practicability of similar bolt circles and number of bolts.

The Committee being desirous of recommending a standard that would be endorsed and adopted by the manufacturers, feels much pleasure in communicating with you and asking your coöperation. We would, therefore, be obliged if you will furnish us with the following information :

1. A catalogue or list giving the flange diameters, diameter of bolt circle, and size and number of bolts you have adopted in your works ?

2. What special reasons had you for the adoption of the sizes you now use ?

3. Would you be willing to modify those standards in your future work to make them conform to a uniform one ?

The divergencies now known to exist are so trifling that we do not think the changes we might ask you to make will be very expensive or troublesome. The Committee will appreciate any advice or suggestion you may feel disposed to offer, and hope to have an early reply.

Yours respectfully,

PERCY A. SANGUINETTI,
Chairman.
Franklin Sugar Refinery, Philadelphia.

The other members of the Committee to whom replies may be sent are :
E. F. C. DAVIS, P. & R. Coal & Iron Co., Pottsville, Pa.

W. F. MATTES, Chief Engineer of West Superior Iron & Steel Co., W. Sup., Wis.

A. H. RAYNAL, Richmond Locomotive and Machine Works, Richmond, Va.

S. S. WEBBER, Ass't Manager of Trenton Iron Co., Trenton, N. J.

On motion, the Report was accepted and the Committee continued.

A letter was read to the Convention, from the members of the Society resident in the City of Richmond, Va., urging most cordially upon the members that their city should be selected by the Council for the eleventh annual meeting in the autumn of this year. The matter of choosing the date and place of the annual meeting being, under the rules, in the hands of the Council, the letter was referred to them, with the statement that there was every probability that the invitation would be accepted.

The following resolution was offered by Mr. Wm. Forsyth, and seconded by Mr. D. L. Barnes:

"*Resolved*, That a Committee of Seven be appointed by the Chair, to report on Standard Methods of Conducting Tests of Efficiency of Locomotives, including the engine, the boiler, the quality of the steam, and the comparative efficiencies of simple and compound locomotives."

The debate on this resolution, was as follows:

The President.—While I myself favor a very full amount of business of the kind by the Society—investigating all sorts o engineering subjects and having Committees to report on them, we must, of course, be careful that we do not have too many Committees running at once, and before we vote I think there should be some discussion of the matter.

Mr. E. B. Wall.—I think this is a very good resolution and I hope it will prevail. The appointment of a Committee on this subject would afford an opportunity for an exchange of views between engineers engaged in the various fields of steam engineering. Those interested in the locomotive would be benefited by the independent, and probably new, opinions advanced by engineers devoted to marine and stationary engineering, and these latter gentlemen would probably find after their investigation and discussion that the locomotive design was not so inferior as is sometimes hinted. I therefore hope that some o the members appointed on the Committee will be men not connected with locomotive engineering.

Mr. W. O. Webber.—I would like to second the remarks made

by the gentleman who just sat down, because, from my own experience during some three or four years on the Chicago, Burlington and Quincy Road testing locomotives, I found almost no original data to refer to in any way, and it seems to me that with the number of roads now having test departments looking into the efficiency of locomotives and making valuable tests, that a great deal of their work and data is necessarily almost useless, you might say, because the tests made on one road cannot be readily compared with the tests made on another. And, as I understand it, this Committee's work would be largely to unify that matter so that the tests made on one road would be comparable with the test made on another. I very earnestly hope that this motion will prevail; and that this Committee will be appointed.

Mr. Geo. S. Strong.—I should like very much to have something of this kind done. I think the time is not so far distant when the question of economical working of locomotives is going to be of as great interest to railroad companies as the question of economy of stationary engines has been to manufacturers. As Mr. Webber has remarked, there has been very little done up to this time in the way of testing locomotives in such a way as to make the test reliable, and when one makes an attempt in that direction he is generally met by the difficulty of having some man appointed to co-operate with him who has his own ideas, and in addition to that man's ideas not being in accordance with the best engineering practice, perhaps he finds that a very large proportion of the people he has to deal with in the operation of the line try to arrange matters in such a way as to invalidate the facts in the case. And if we had some standard method of conducting these tests, so that in case a test was made, it should be conducted according to the standard of the mechanical engineers, it would do away with all questions of this kind, and it would establish some fixed method of doing these things which would do away with any opportunity for trouble between an expert who is trying to make a test and the man who is appointed by a railroad company to coöperate with him in the test. I have spent several thousand dollars in conducting locomotive tests and I have found that difficulty. Where I employed a competent expert, perhaps some man who did not know anything about making a test of a steam-engine at all would be appointed to coöperate with him in making the test ;

and that man would raise some question of the method pursued which probably would invalidate the whole thing after it was done. As every one knows, a man in an official position with a railroad company has more influence with the public than an expert, no matter what that expert's reputation for honesty and capability may be. When he goes before the railroad public and is decried by a man who is an official of the railroad company, that expert's testimony is of no value with the railroad company whatever. It simply is invalidated by what some man may say who has no reputation as an engineer.

Mr. D. L. Barnes.—It would seem that the American Society of Mechanical Engineers ought to pay as much attention to locomotive engineering as stationary engineering. There is as much coal used in the production of mechanical horse-power in locomotive engines in this country as in stationary engines, and locomotives require as much attention and should have it. The locomotive engine is not a small engine. It is an engine varying from 200 to 1,000 horse-power. The reason why we have very little, if any, reliable information regarding the locomotive to-day is because the trials are very difficult to make, and it requires an expert, and a well-informed one, to make a test which is satisfactorily comparable with other tests of locomotives and stationary engines. In the coming exposition, to be known as the Columbian Exposition, there will probably be competitive trials of locomotives. An endeavor will be made to obtain foreign competition; and it is to be hoped that before that time this Committee which is about to be appointed will be enabled to report, giving us a satisfactory method of making tests. The railroad companies in this country are putting out considerable sums of money every year testing locomotives, and some conclusions are reached. It is not for lack of money that these tests are not more satisfactory, but because of lack of proper methods. Probably one of the most difficult points to determine is the character of the steam entering the locomotive cylinders. It is wet, beyond question. A satisfactory calorimeter is a most difficult problem; and I would like to hear expressions from members here, if this discussion continues further, regarding such a calorimeter, because upon some tests soon to be made I would like to apply a satisfactory device.

The motion for the appointment of a Committee was then put and carried.

The President.—I will not promise in appointing the Committee to carry out the suggestion not to appoint any locomotive engineers upon it. I think if we do not have some one on the Committee who knows something about the subject we shall not get a very valuable report.

Mr. Wall.—I would like to make a little correction there. I did not want this Committee made up entirely of men who were not locomotive men. I wanted a few of them to be steam engineers who have no particular interest in locomotives.

The President.—The right way to make such a Committee as that is to put some men on who are specialists in locomotives, and other men who are not, but who are familiar with engines in general.

The motion to appoint a Committee, being put, was carried, and the President subsequently appointed as such Committee: Prof. Chas. B. Richards, of New Haven; Mr. Wm. Forsyth, of Aurora, Ill.; Mr. H. B. Stone, of Chicago, Ill.; Prof. Jas. E. Denton, of Hoboken, N. J.; Mr. F. W. Dean, of Boston; Mr. Axel Vogt, of Altoona, Pa.; Mr. Allan Stirling, of New York.

At a later point in the session, was also presented the report of the Society's Committee appointed to express to the Mayors of Boston, Chicago, St. Louis, and Washington the desire of the Society that a world's fair should be held in 1892. The report and the replies thereto, were as follows:

NEW YORK, *Dec.* 4th, 1889.

TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS:

Your Committee, appointed to lay before the municipal authorities the vote of the Society in reference to the holding of a world's fair in 1892, would respectfully report that they have attended to the matter placed in their charge by calling upon Mayor Grant in person and presenting the letter of which the enclosed is a copy; and they have also forwarded duplicates of said letter, excepting only the address, to the Mayors of Boston, Chicago, and St. Louis, and the Chief Commissioner of Washington City.

All of which is respectfully submitted.

GEO. H. BABCOCK,
HENRY MORTON, } *Committee.*
J. F. HOLLOWAY,

NEW YORK, *Dec.* 3d, 1889.

The American Society of Mechanical Engineers, at a recent Convention held in the City of New York, appointed the undersigned a Committee to express to you their deep interest in and sympathy with the proposed holding of an international exposition, in commemoration of the four hundredth anniversary of the discovery of America by Christopher Columbus.

As the Society is a national one, the individual members necessarily have their preferences as to the location of the exposition, but the Society as such directs us to assure you of their united and hearty coöperation in any and every way that will best promote the end in view, and at whatever time or place that may be selected, be it Boston, Chicago, New York, St. Louis, or Washington.

Very truly yours,

(SIGNED) *Committee.*

TO THE HON. MAYOR OF ——

(Addressed.)

TIMES BUILDING, NEW YORK, *Dec.* 9th, 1889.

DEAR SIRS:

The Mayor directs me to acknowledge the receipt of your communication of December the third containing an expression of your deep interest in and sympathy with the proposed international exposition of 1892.

The Mayor and the General Committee direct me to thank you, and through you the American Society of Mechanical Engineers, for your kind interest in the exposition.

Respectfully,

W. McM. SPEER,
Secretary.

GEO. H. BABCOCK, Esq're,
HENRY MORTON, Esq're,
J. F. HOLLOWAY, Esq're,
} *Committee.*

CHICAGO, *Dec.* 13th, 1889.

GEO. A. BABCOCK, HENRY MORTON, J. F. HOLLOWAY, 30 Cortlandt St., New York City.

GENTLEMEN:

Your communication of the 3d inst. to the Mayor has been received, and I am instructed by him to acknowledge same. Your view is one which, we think, is held by all of our people. While we hope and expect that the Exposition will be held in Chicago, it will receive our hearty and cordial support if held in any other city.

Yours truly,

E. F. CRAGIN,
Secretary.

WASHINGTON, *Dec.* 11th, 1889.

MESSRS. GEO. H. BABCOCK AND OTHERS, COMMITTEE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 30 Cortlandt St., New York, N. Y.

GENTLEMEN:

The Commissioners of the District of Columbia, who are the municipal authority at the seat of government of the United States, direct me to acknowledge receipt of yours of the 3d inst., addressed to the Mayor of Washington, expressing the sympathy of your Society with the proposed project of holding an international exhibition in commemoration of the four hundredth anniversary of the discovery of America by Christopher Columbus, and giving assurance of the hearty coöperation of your Society.

Respectfully,

W. TINDALL,
Secretary.

The report was accepted, and with the completion of its duty, the Committee was discharged.

Mr. Jas. W. See, Chairman of the Society's Committee on Securing a Governmental Bureau of Standards, reported as follows:

REPORT OF COMMITTEE ON GOVERNMENTAL BUREAU OF STANDARDS.

Your Committee would report:

1st. That a bill has been drafted and presented to the Lower House of Congress.

2d. That the bill was referred to the Committee on Patents.

3d. That your Committee has appeared before the Committee on Patents and urged the merits of the bill.

4th. That a favorable report to the House is promised.

5th. That the future of the bill cannot be foretold.

6th. That further time is requested.

JAMES W. SEE,
for Committee.

CINCINNATI, *May* 16th, 1890.

The professional papers were then taken up as follows:

That by Mr. W. F. Dixon, on "The Efficiency of Locomotives," was discussed by Messrs. W. E. Hall, Angus Sinclair, W. O. Webber, Geo. S. Strong, D. L. Barnes, L. S. Wright, W. F. Durfee, and A. T. Woods.

The second paper was in a similar line, entitled, "Working Railways by Electricity," by Mr. Willis E. Hall. The discussion was by Messrs. Scheffler, Oberlin Smith, H. C. Spalding, McFarren, E. P. Roberts, Jesse M. Smith, Barnaby, Barnes, Sweeney, Raynal, Forsyth, Rogers, Dashiell, W. O. Webber, and T. R. Morgan, Sr.

The paper by Mr. A. F. Nagle, entitled, "Determination of Sensitiveness of Automatic Sprinklers," was read and discussed by Prof. D. S. Jacobus before adjournment, but the session adjourned in the middle of the debate.

THIRD SESSION, WEDNESDAY, MAY 14TH.

The third session was called to order at 2:30, and the debate on Mr. Nagle's paper was continued by discussions by Messrs. Emery, Thurston, and Woodbury.

Prof. R. C. Carpenter presented his three papers: "Tests of Several Types of Engines under Conditions found in Actual

Practice," "Comparative Tests of Hot-water and Steam-heating Plant," and his "Note on Kerosene in Steam-boiler."

The first was discussed by Messrs. Jacobus, Borden, Nagle, and Webber; the second by Messrs. Emery, Gobeille, and Robb; the third by Messrs. Porterfield, Nason, and Nagle.

The rainy afternoon precluded the projected drive which was a feature of the programme for the afternoon, and it was transposed to Friday. The papers allotted to the Friday afternoon session were taken up. Three of these papers having relation to the subject of steam-engine governing, they were read and discussed together, as follows: E. J. Armstrong, "The Use for Inertia in Shaft Governors," Jesse M. Smith, "Governor for Steam-engines," John E. Sweet, "Effect of an Unbalanced Eccentric in Shaft-governed Engines." The discussion was participated in by Messrs. Armstrong, Sweet, Thurston, Fawcett, Jesse M. Smith, C. M. W. Smith, Webb, Dutton, Robb, Jacobus, and Barnaby.

Prof. Jacobus discussed Mr. W. W. Bird's paper, "An Open-end Mercury Column for High Pressures;" and the list of papers was concluded by that of Prof. G. I. Alden, entitled "Automatic Absorption Dynamomometer." Topical questions were discussed until adjournment. The first was "Does a boiler steam more freely if the tubes are arranged so as to be farther apart horizontally in the upper row than in the lower rows?" Messrs. Sweeney, Cole, Suplee, Dutton, Nagle, Barnes, and Webb took part. Messrs. Barnes, Raynal, and Penney spoke as to the greatest number of times per minute that a dash-pot could be lifted, such as used in the Corliss valve-gear. Messrs. See, Oberlin Smith, and Crocker spoke of the advantages as to the construction and operation offered by vertical bending rolls as compared with horizontal ones. Messrs. Fawcett, Laird, and Suplee gave expressions as to the purification of bad feed-water for boilers. The session then adjourned.

The fourth session was opened that same evening, at the same place at 8 o'clock.

Prof. J. Burkitt Webb presented his two papers on "Chimney Draft," which were discussed by Messrs. Gale, Gray, Nagle, Magruder, Borden, and Barrus.

Two papers on "Chimney Draft," one by Prof. Wood, and the other by Mr. Kent, were read at the same time, and were discussed by Prof. Gale and Messrs. Nagle and Scheffler.

Prof. Wood's papers, "Graphic Representation of Thermo-Dynamic Quantities," "Test of a Refrigerating Plant," and Prof. Thurston's paper, "Hirn and Dwelshauvers' Theory of the Real Steam-engine," received no discussion.

The paper by Geo. H. Barrus, "A Universal Steam Calorimeter," was discussed by Messrs. Robb, Barnes, and Spangler.

Messrs. Henning, Samuels, Wright, Nason, Raynal, Hibbard, and C. M. W. Smith discussed the best form of hydraulic valves.

Messrs. Penney and Nason discussed the use of power moulding machines in foundries.

FRIDAY, MAY 16TH.

The fifth and concluding session was called to order in the Scottish Rite Cathedral, at 10 o'clock. The President announced the Committee to nominate officers of the Society for the ensuing year, under Article XXXI. of the Rules, to be Messrs. C. W. Hunt, of New York; Henry G. Morris, of Philadelphia; B. H. Warren, of Boston; G. A. Gray, of Cincinnati; and Jesse M. Smith, of Detroit.

Mr. D. K. Nicholson's paper on "Heating Furnaces" received no discussion. Mr. Suplee's paper on "Equilibrium Arch Curves" was discussed by Messrs. Nagle, Landreth, and Webb. The paper by T. C. Clarke, entitled, "The Kinzua Viaduct," received no discussion. Messrs. Barnaby and Suplee discussed Prof. Webb's paper on "The Length of an Indicator Card." Prof. Webb then introduced the following resolutions:

"*Resolved*, That the Society deeply regrets the necessity which has compelled the absence of Secretary F. R. Hutton, whose presence and labors have always contributed so largely to the success of the meetings of the Society."

"*Resolved*, That the members in attendance during the various sessions of the Society during the Cincinnati meeting desire to express to Mr. C. J. H. Woodbury, Acting Secretary of the meeting, their full appreciation of the courteous and efficient manner in which he has filled the place of the absent Secretary."

Prof. D. S. Jacobus then presented his two papers, the first "Indicator-Cards of the Pawtucket Engine," and second, the "Effective Area of Propeller Screws." Messrs. Nagle, Suplee, Raynal, Barnaby, Webb, J. W. See, and Ashworth discussed the first, and Messrs. Emery, Raynal, Oberlin, Smith, Henning, Cole, and Suplee discussed the second.

Mr. Woodbury then moved the following resolution of thanks

to Mr. Bond, who had with him shared the burden of attending to the details of the meeting, and which was presented as follows:

"*Resolved*, That the thanks of this Society are hereby tendered to Mr. Geo. M. Bond for his valuable assistance to the Acting Secretary of the meeting."

Mr. Ashworth brought up the question of an Organization of Engineering Societies, and the following remarks were made:

Mr. Daniel Ashworth.—Mr. President, under this general tenor of business I would like to ask for some information regarding a matter which was passed upon at the New York meeting and upon which you touched yesterday. Is there not in the hands of a Committee a decision or a report as to the advisability of creating a separate organization composed of the most prominent members? Am I correct in that?

The President.—Shall I state, Mr. Ashworth, briefly, just how the case stands?

Mr. Ashworth.—Yes, sir; if you please.

The President.—There was a Committee appointed of three members, Mr. Henry R. Towne, Professor Thurston, and Dr. Sellers, at the autumn meeting of this Society, to consult and confer with other Committees from the three other national Societies, the Civil Engineers, the Mining Engineers, and the Electrical Engineers—each of those Committees consisting of three well-known members of their Society. That Joint Committee of twelve is now working upon the scheme of establishing in the near future, not the mingling of the present societies into one society, nor a special society which is entirely outside of this, but a sort of *Federation* of all the present Societies, including these four, with possibly the Institute of Architects, the Naval Engineers' Society, and other associations ranking with us which may be started in the future. This Federation would be representative of *all* American Engineers, of *all* classes, in the different departments, Civil, Mining, Mechanical, Electrical, Naval, Military, Sanitary, Architectural, etc. This, our own Society, would be connected with it, although not merged into it in any sense whatever. The local Societies, the small Societies, which are scattered all over the country, which now have but little connection with each other, and feel that they have no general system of working, would, perhaps, be Chapters of this new organization.

There has, I know, been some feeling in this Society that if

the proposed new one was started it would reduce our dignity and usefulness. I think this is entirely a mistake, and that when the new scheme is perfected and put properly before our members, so that they can understand it, they will say that there is nothing in it in any way derogatory to the present societies, but only that which will help them in their work and enable them to pull together better and give a better tone to engineering over the whole land, in our own eyes and in the eyes of foreigners. It was brought up at this time because it was thought that the great "Institution of Civil Engineers" of England might be invited over here in 1893, and there is now nobody in particular to entertain them as they entertained us. That is what started the matter, but I think that without such comparatively minor reasons, our different Societies and individual engineers should have some common method of coming together and getting at each other's transactions easily, and some way of having a common library, so that the great expense of duplicating models, pictures, and books would not be incurred as it is now. Such a federation would mean simply that they should work on a system, instead of a total lack of system—instead of rivalling each other as they do now.

I think the danger of there being in such an association any ring or cliqueism, which seems to be feared, can be entirely done away with by the manner of organization and the method by which the members are elected. This can be arranged in such a way that all the members of all the Societies will have a full and free chance to vote as to who is worthy for the higher places in the new Society, should there be different grades of membership. It is possible that it may be started, if started at all, somewhat on the plan of the great foreign scientific Societies, with a higher grade of "fellowships," but I do not know how that is going to be, as it is all in the hands of the Committee. When they get the thing in proper shape and a joint report is made out, it will undoubtedly be distributed to all the members of this and other Societies—giving them a chance to vote on the subject and say whether it is desirable to go forward. All that I know of the present status of the scheme comes from the Committee's informal report which was submitted to our Council at its last meeting in New York—simply reporting progress. Undoubtedly it will take several months yet to get the thing into any definite form.

Mr. Ashworth.—I have taken a deep interest in this matter, and in the absence of a report from that Committee, even reporting progress, I was congratulating myself, and so were many of the members, that it had been sent to some inaccessible pigeon-hole, for I still think that it has with it an un-American idea. It is undemocratic, and I still believe it would create a spirit of caste—a special class. You might just as well strike from the name of this Society the word American as to take such an action. I am satisfied that it will be an entering wedge for destroying the efficiency of this organization. Why not pursue the ordinary course if we are to coöperate with our sister Societies as we should, and have Committees appointed, and act in conjunction with them for several or for special purposes. These grades of fellowship that you have touched upon are in the same line. There will be no end to that. I am in favor firmly of the American Society of Mechanical Engineers intact as it is, and opposed to a confederation, and I believe that I voice the sentiments of the members present.

Mr. Webber discussed the question as to the desirability of having the lead increase with the load in high-speed automatic engines.

The following resolutions of thanks were presented, duly seconded, and carried with enthusiasm, and the professional sessions adjourned.

Resolved, That the thanks of the Society are extended to the local Committee of Arrangements for the Cincinnati meeting.

The pleasant features and general success of the meeting have been brought about largely by the efforts of these gentlemen in securing entertainment for the visiting members and guests, sparing no pains to further the object of the meeting in all respects, and making it one of which we shall carry with us most pleasant recollections. The members would in particular express their thanks and recognition to Gen. Sir A. T. Goshorn and the Directors of the Cincinnati Art Museum for their abundant courtesy and hospitality, and for the opportunity to inspect the rich treasures there gathered from every source.

Resolved, That the thanks of the Society be tendered on behalf of the ladies accompanying the members, for their entertainment by the local committee of ladies. Much pleasure has been derived from their various excursions and drives about the city, to the Rockwood Pottery and other places of special interest to them, and the visits of the ladies to Cincinnati. The excursion day of our meeting is one which all who were privileged to attend will remember with the greatest of pleasure, and the acknowledgment of the hospitality which made the day so delightful is an equal pleasure.

Resolved, That the American Society of Mechanical Engineers express to The Niles Tool Works its fullest appreciation of the hospitality so warmly ex-

tended, both in visiting the extensive works and also in partaking of the bountiful lunch at Woodsdale Park.

Resolved, That the thanks of the Society be extended in the warmest manner to Messrs. Proctor and Gamble, for the opportunity afforded of visiting the extensive and admirably arranged works at Ivorydale.

Resolved, That the thanks of the Society be tendered to the Ohio Mechanics' Institute ; to the Cincinnati Chamber of Commerce ; to the Triumph Compound Engine Co.; to the Cincinnati Gas Light and Coke Co.; and other educational, commercial, and manufacturing institutions, which have thrown open their doors to the members, with invitations to freely visit them. These invitations have so far as possible been made use of by members, and have contributed much to the pleasure and benefits of our visit to the city.

Resolved, That the American Society of Mechanical Engineers expresses its appreciation of the excellence of the transportation facilities extended to the Eastern members by the P. R. R. Co., and that thanks are due to this company and to Mr. Samuel Carpenter, Eastern Passenger Agent, and Mr. H. M. Haines, General Passenger Agent of the Company. Those who enjoyed the advantages offered by the P. R. R. Co.'s special train from New York to Cincinnati will remember the trip as a thoroughly entertaining preliminary to the meeting.

Resolved, Also, that the Society is indebted to the Cincinnati, Hamilton and Dayton R. R. Co. for the special train kindly furnished on the occasion of the excursion to the works of Messrs. Proctor & Gamble, and to The Niles Tool Works, and for the freedom of Woodsdale Park.

Resolved, That the thanks of the American Society of Mechanical Engineers are due to Mr. Ralph Peters and the Managers of the Railways running out of Cincinnati, which afforded the Society an opportunity to pass over two of the bridges crossing the Ohio and to have views of the other two bridges, all of which are of especial interest to engineers, as being at the time of their erection the longest spans of the kind in the world. Furthermore, we appreciate the honor of being on the first passenger train entering the city at the new station.

In addition to the hospitality which we have already received, we have also to express in anticipation the pleasure which we feel in accepting the invitation which has been extended to the Society by the Addyston Pipe Works.

Therefore :

Resolved, That the appreciative thanks of the American Society of Mechanical Engineers be extended to the Addyston Pipe Works for their warm invitation to visit their establishment, and to Mr. Ralph Peters, for extending special train facilities for this visit.

Entertainments and Excursions.

Besides the reception given to members and their guests, in the Hall of the Scottish Rite, on Tuesday evening, the hosts of the Society, in Cincinnati, entertained them by an excursion on Thursday, May 15th, to Hamilton.

A special train was put at the services of the Society, by the courtesy of the C. H. & D. R. R., and was drawn by a Strong loco-

motive. The Niles Tool Works, at Hamilton, were thrown open for inspection, and after the visit they were conveyed to Woodsdale Island, some five miles from Hamilton, where the same firm entertained its guests at luncheon. After the banquet, a stroll over the grounds, and other entertainments, the party returned to Ivorydale, where they spent over an hour after-time in a visit to the great soap works of Messrs. Proctor & Gamble.

In the evening of this day, the Cincinnati Art Museum, with its treasures of interest and value, was thrown open for a reception to the engineers. A speech of welcome was made by Gen. Sir A. T. Goshorn, to which Mr. C. J. H. Woodbury responded in the absence of the President. The evening was a most enjoyable one.

On Friday afternoon, the Penna. R. R. Co., through the courtesy of Mr. Ralph Peters, gave a special train excursion over the railway bridges at Cincinnati, and at the end of the bridge, carriages were in waiting to take the party around the beautiful suburbs of the city. The drive was enjoyed for something over three hours.

CCCLXXX.

REPORT OF COMMITTEE ON STANDARD TESTS AND METHODS OF TESTING MATERIALS.

Mr. Chairman, and Gentlemen of the
American Society of Mechanical Engineers:

Herewith your Committee appointed to consider "Tests and Methods of Testing," beg to submit the following

Report.

The Committee having been appointed by the Chairman, held several preliminary meetings, in which a general plan of procedure was discussed and accepted.

It was decided to send to all of the principal parties interested in testing, duplicate samples of one grade of material for test purposes. These parties were requested to send in their reports of tests of this material in full.

Upon receipt of such reports, the Committee examined them carefully in order to arrive at a comparison, and try to formulate a general method of procedure for future tests. It was, however, found to be impossible to formulate or tabulate these results, and that even a personal acquaintance with all of the parties making these reports made it difficult to interpret the exact meaning of all the reports.

It required a table of 32 columns to record all of the results reported, and this made it impossible to arrive at any conclusion or general deductions.

Then the Committee proposed to send out another set of test pieces, containing duplicate and triplicate samples, and also a printed blank for the purpose of reporting the results of tests.

This printed blank suggested by your Committee is shown annexed to Report as Table I., Addendum 1.

A circular letter was therefore issued in August, 1886, as given below, and sent to 24 different parties, with 236 test pieces divided among them, and as follows:

Form 9–58, '86. (*August*, '86.)

The American Society of Mechanical Engineers, at a recent meeting, appointed a Committee to see if there could be recommended a standard form of test specimen and a standard method of test for engineers' use.

By such standards, it would be possible for the tests of different experimenters on the same material to be comparable, and much confusion, misunderstanding, and duplication would be avoided.

As a step in the labor of that Committee there are sent to you prepaid by express the following samples for test, viz.:

Pieces round steel, $\frac{3}{4}$ inch diameter.
Pieces round iron, $\frac{3}{4}$ inch diameter.
Pieces flat or square steel.
Pieces flat or square iron.

We hope it will be convenient for you to coöperate with this Committee by testing the above specimens in your testing machine, and to report the results obtained upon the enclosed blank—which we have prepared for that purpose in order that all of the tests made for the Committee may be readily and accurately compiled.

We desire that the testing should be done in your regular manner, with neither more or less care than such work is ordinarily done in the various testing laboratories of the country, and in this way to determine how it may be best, under existing methods, to collate and compare the work of different experimenters.

We have made arrangements by which similar tests will be made by numerous other parties, the material furnished to all being as nearly identical as could be obtained by special care and selection. The result of the tests when obtained will be carefully tabulated by the Committee and then embodied in a report to the Society, copies of which we will be glad to send you when complete.

Should you see fit, in view of their scientific and practical value, to make these tests without expense to the Committee, your liberality will be highly appreciated, and duly acknowledged, but if not, we still wish to have them made and will promptly pay all reasonable charges therefor out of a small fund which has been contributed for our work.

The specimens are to be tested in tension only, but with obser-

vations which will show both their elastic limit and ultimate strength.

In making the tests, such observations of the specimen and of the testing apparatus should be made as will give all the items of information called for on the blank form of report.

The observations of elongation should be noted with your regular measuring or length gauging apparatus, if you have one. If not, by making five prick marks on the specimen 8 inches apart, and correctly measuring the separation of these parts with a micrometer or a fine scale, before, during, and after the test. In all cases the gauge length should conform to the standard of 8 inches as closely as possible. It is desired that each test should be made with as much rapidity as is consistent with accurate work, and that the actual time so occupied be recorded in the proper column.

When returning the report we will be glad if you will add to it any further particulars regarding your testing machine—additional to those called for on the blank—which will identify it, such as date of design or patent, makers, number, size, or any other designating mark by which it can be distinguished. Also a brief description of the kind of holders employed and the means used for measuring the elongation, and any other particulars relating either to the machine or to the mode of making the tests. The specimens after testing should also be returned, properly boxed, to the address of the Society as above, care being taken to mark them so that they can be identified as having been tested by you. The report of tests should be sent by mail, addressed to this Committee at Society's Headquarters. Trusting that you will coöperate with us in this important and interesting work assigned to the Committee, we are, respectfully,

HENRY R. TOWNE,
GUS. C. HENNING,
R. H. THURSTON,
CHAS. H. MORGAN,
DR. THOS. EGLESTON, } *Committee.*

NAME.	ADDRESS.	NO. OF P'C'S.
Purdue University	Lafayette, Ind......A. W. Stahl, U. S. N.	9
Cornell University	Ithaca, N. Y....R. H. Thurston, Director.	9
Rensselaer Polytechnic	Troy, N. Y.	4
Stevens Institute	Hoboken, N. J ....J. E. Denton.	9
Union Bridge Co	Athens, Pa...............Chas. Kellogg.	12
Edgemoor Iron Co	Wilmington, Del.....Wm. Sellers, Prest.	12
Union Iron Mills	Pittsburgh, Pa..Carnegie Bros. & Co., Limited.	12
Pencoyd Iron Works	Pencoyd, Pa...........A. and P. Roberts.	12
Cambria Iron Co	Johnstown, Pa............C. A. Marshall.	12
Trenton Iron Co	Trenton, N. J ....Wm. Hewitt.	9
Pennsylvania Steel Co	Steelton, Pa.............Edgar C. Felton.	9
Edgar Thompson Steel Works	Bessemer, Pa..Carnegie Bros. & Co., Limited.	10
Midvale Steel Co	Nicetown, Phila., Pa..Wm. Sellers, Prest.	9
Pottsville Steel and Iron Co	Pottsville, Pa..........Atkins Bros. & Co.	8
Fairbanks Testing Laboratory	34 Thomas St., N. Y........A. V. Abbott.	10
Riehlé Bros. & Co	Philadelphia, Pa.	12
Tinius Olsen & Co	Philadelphia, Pa.	12
Pittsburgh Testing Laboratory	98 4th Ave., Pittsburgh, Pa., Hunt&Clapp.	12
Watertown Arsenal	Watertown, N. Y., Col. F. H. Parker, Com.	12
School of Mines	Columbia College, 49th St. and 4th Ave., F. R. Hutton.	6
Phœnix Iron Co	Phœnixville, Pa ....D. H. Reeves.	6
South Chester Rolling Mill Co	Thurlow, Pa.............C. B. Houston.	12
Yale & Towne M'f'g Co	Stamford, Conn......H. R. Towne, Prest.	12
Washburn & Moen M'f'g Co	Worcester, Mass...........C. H. Morgan.	6

Of these 24 parties, two reported inability to test, as machines were unsuitable; two parties declined; one reports specimens gone astray; one party reports having made tests, but that the report was lost in the mail; one party is still engaged in making the tests and promises to report results; and sixteen parties have sent in complete reports, as given in Addendum 1, Tables II. *a—t*.

The Committee would particularly express their appreciation of the work of the late Mr. Chas. A. Marshall, C.E., Engineer of Tests Cambria Iron Co. (who unfortunately was one of the victims of the Johnstown disaster), for the masterly reports sent in and the pains he took in making these tests.

The Committee also desires to express their gratification and obligations to the courteous and willing assistance offered them on all sides, and for the care taken in making the tests and for the completeness of the reports returned.

It became apparent to your Committee that it would be impossible to arrive at any general results or be able to report in a reasonable time if the recommendations to be formulated were to

be deduced from long series of experiments made by it. Moreover, it appeared desirable to propose International Standards, if this were possible, and with this in view one member of your Committee, during two years spent abroad, had frequent consultations with the most prominent engineers interested in the matter and obtained information on the subject which has shown how useless it would have been to repeat the work of other able experimenters, whose work has always been considered above criticism. The cost of such series of tests would also have been a serious obstacle.

It was ascertained that in *France* no uniform method or standards had been proposed or adopted, except that the Government, making the use of a Thomasset or Maillard machine compulsory, had adopted a very small test piece, approximately the same as the Woolwich Standard.

General testing is not carried out according to a uniform method; in fact, most engineers followed their own inclination, which is governed by the particular testing machine at command. The use of testing machines is very limited in France except for Government work.

As the French Government makes the use of a Thomasset or Maillard machine in all its work compulsory, the introduction of other forms has been almost impossible, and the development of this class of machines is at a standstill.

Particular attention is called to the fact that this does not apply to that part of old France now belonging to Germany, but to the present territory of *France*.

In *Great Britain*, the use of testing machines is much more general and the types of machines used are various in form, and capacity generally being ponderous. The methods of testing have not been reduced to a standard proceeding, and the results obtained are as a rule scantily reported, so as to be almost useless for comparative purposes, and the observations taken are few and incomplete.

The leading engineers, however, more particularly those in charge of laboratories connected with schools and colleges, execute their work in a very accurate and complete manner, and the shapes of test-pieces and methods of testing are very nearly alike.

In *Belgium*, testing is more general than in France, but although machines are more numerous and more diversified in type, there

is as yet neither general method adopted or proposed, nor a standard form of test pieces in general use.

In *Germany*, including the annexed provinces, testing, in general, is more advanced and general than in any other country, and almost all types of machines are in use, while the form of test pieces and method of procedure are quite uniform and identical with those adopted in *Austria*, *Switzerland*, *Russia*, and *Denmark*.

During the last decade German manufacturers, engineers, and professors, in conjunction with their fellows in Austria, Switzerland, and Russia, have had a number of largely attended conferences in which a highly successful effort was made to adopt standard methods, test pieces, and machines.

The standards proposed at these conferences have been practically adopted almost universally in those five countries.

The general results and deliberations of these conferences are given in full in Addendum 2, containing also a list of the names of participants. These conferences proposed a great amount of original investigation to determine doubtful points and matters in debate, and as soon as such series of tests had been completed a new conference was called to deliberate upon the reports submitted. Several series of investigations were delegated to different investigators or committees, and a vast amount of work accomplished, without any expense to individuals, the results obtained being all published in full in the two most important technical publications in that branch, namely, the "Mittheilungen aus der Versuchs Anstalt des k. Polytechnikum Muenchen," Prof. J. Bauschinger, and "Mittheilungen aus der k.k. Versuchs Anstalt Charlottenburg," Prof. Martens. None of the recommendations of these conferences were to be binding or final, inasmuch as new light thrown upon this subject might make a change seem desirable, but the immediate adoption of them was strongly advocated until such changes became desirable.

As a result, these recommendations have been introduced universally and have proved to be of incalculable benefit to all parties interested. A perusal of the report of these conferences will show that the recommendations advocated are in all essential particulars identical with custom in Great Britain as well as the United States, and with slight modifications to suit particular cases, similar standards could be proposed in all countries so as to make them international.

Several of the paragraphs refer to tests which have, until the

present time, been rarely made in this country, and in order to indicate the method of carrying them out properly, the German methods have been adhered to. Thus drop tests are not often made, and then only with the crudest apparatus, never giving reliable results, and making it impossible to take accurate measurements.

As it is only a question of time when such tests will be made regularly here, it was thought advisable to recommend the adoption of apparatus and methods which are known from long experience to give satisfactory results, and also to propose the adoption of such apparatus and methods as have been in common use elsewhere and have given valuable results, and which would make comparisons easy and also avoid changing later on, should an international convention become a fact.

The same is to be said about some tests of stone, which in this country are never carried out; *i.e.*, the freezing, quarrying, and abrasion tests. These have been carried out on an extensive scale in France and Germany by the highest authorities, and it would not benefit any one here to adopt different methods; it would on the contrary merely produce confusion to adopt new methods now, and those suggested in Europe are given in Appendix 2.

Moreover, comparisons would be impossible and, as all testing is merely comparative, the principal value of work otherwise highly desirable would be lost.

Testing Machine.—It is not deemed advisable to recommend the use or adoption of any particular form of testing machine, as there are a number now in daily use which, when kept in good repair, can be made to give good results.

Shackles and Holders.—The use of the form of wedge holders in almost universal use, however, can be hardly too strongly deprecated, giving invariably lower results than the material would show with a good form of holders.

Abrasion Tests.—No recommendations as to abrasion tests of straight or curved rails can as yet be offered; the subject has as yet been so little investigated that much work must be done before any methods or apparatus can be adopted.

It is, however, undeniable that rails should be tested to determine their wearing qualities under the influence of impact and friction under high speeds; also the relative effects under wide ranges of temperature and moisture. Also, as nearly nine-tenths of the rails used go into the track under permanent set due to curving, rails should be tested principally in such condition. Before

such comprehensive tests are made regularly, very little can be said in advance of the prospective value or wearing properties of rails. The relation between tenacity, elasticity or resilience and the life of rails, has never yet been determined in the remotest degree.

Again, the effect of different kinds of tires is another field of investigation which must largely affect the wear and life of rails, and require special investigations not yet attempted. Neither rails nor axles need be subjected to bending tests by static loads, as they are invariably strong enough in that respect, and there is no question about their strength or quality after the drop test, which is critical. has been applied.

Similarly, the hammer test for tires can be dispensed with, as no additional information is obtained thereby after the drop test has been made.

Piece Tests.—Up to the present time no system of testing has been proposed which would give a good knowledge of a large lot of pieces. It is plain that when rods, rails, axles, and the like are rolled in quantity, the examination of one or two samples per ton of finished product does not give satisfactory information except in a general way; and, although it seems of great importance to examine each and every piece of any lot of material, all investigations thus far made have given no indications of possible methods to be adopted to this end.

In the fabrication of eye bars for bridge work, the application of a test load equal to about one-third or two-fifths of the load carried within the elastic limit is a useless waste of time, unless strain be measured at the same time. The determination of the coefficient or modulus of elasticity is, however, quite a different matter and may yet lead to valuable results.

Bending the ends of rivet rods is also to be highly recommended, as by this means brittle pieces are easily eliminated; this test, however, proves but one quality and is a very crude one at best.

Giving axles and tires light blows with hand hammers can hardly be considered of value, as such material must indeed be very defective to be affected by such slight impact.

It is highly desirable that cheap and effective methods of making multiple or piece tests be investigated and proposed in order that proper study and discussion of such may lead to standard methods.

In boiler plates it is highly desirable to know the qualities of each and every plate, and methods of testing to determine these

without injuring the material or causing considerable expense and delay are very desirable. When constructing testing machines of all kinds, the possibility of making multiple or piece tests should not be lost sight of.

Such multiple or piece tests have long ago been adopted for finished articles, such as springs, chains, pipes, boilers, cylinders, etc., and a proper method would probably be adopted for other material as well. It must, however, be borne in mind that the application of such tests is combined with considerable difficulty for buyers as well as manufacturers, which might, however, be overcome by proper methods. The results of tests reported to your Committee, and given in *Addendum* 1, *Tables II. a—t*, show the great harmony existing among experts in this country in regard to methods pursued in testing and making out reports. In the latter respect they are, however, not as uniform as seems desirable.

The samples distributed were sent out to all parties alike, without indicating or even intimating to them what to do, except to obtain the results desired as indicated by the several columns on the blank furnished.

It was left to them to obtain the results according to their own methods, and the uniformity of results is satisfactory.

In nearly all respects these tests conform to the recommendations made by your Committee further on, except in regard to the determination of elongations. A study of the new method proposed will probably demonstrate the advisability of adopting the same generally, as it will give results truly comparative and of greater fairness toward the material investigated.

PROPOSED RECOMMENDATIONS FOR STANDARD TESTING.

CONTENTS.

IX. *Shape of Test Pieces.*

1. For tension tests.
2. For compression tests.
3. For transverse tests.
4. For torsion tests.
5. For bending tests.
6. Multiple or piece, welding, hardening, and abrasion tests.

I. General Recommendations.

1. *Testing Machines* must be so arranged that they can be readily and accurately rated, gauged, calibrated, or standardized.

Their construction must be such that stresses can be applied uniformly and without impact, in both hydraulic and screw machines.

For practical purposes it is not necessary that they be automatic in their action.

2. Good *Shackles* or *Holders* must be so constructed that stresses are applied uniformly over the entire section of test piece. Therefore the following requirements become imperative;

α. For *Compression Tests.*

a. Free and easiest possible motion of one of the tables or supports in all directions.

b. The surfaces on which pressure is applied must be as nearly as possible parallel, and must therefore be planed, turned or ground, material permitting.

β. For *Tension Tests.*

a. Free and easy movement of test piece for adjustment at beginning of test.

b. Positive automatic devices which compel the test pieces to assume a truly axial position, before application of stress, at the same time preventing shifting of test piece during test.

These conditions are fulfilled, for round and square and flat pieces, as experience has shown, in a more or less satisfactory manner, by some forms of universal joint with undivided spherical shell, or knife-edge bearings.

For *flat bars* are recommended pin-holes with pin; and with one hole and bolt at each end only, or milled ends and corresponding wedges.

The above conditions are most nearly satisfied by Pummers' Shackles or Holders, or another form as now made by Riehlé Bros., of Philadelphia, and by the Emery Tension Holders.

The use of serrated wedges which cut the test pieces, especially without intermediate adjusting blocks, is to be strongly deprecated.

3. It is not deemed advisable to recommend the adoption of any particular form of testing machine now in use, as a number of well-known types can be made to fulfill the requirements of routine work with sufficient accuracy for practical purposes.

4. *Drop Tests* are to be made on a *Standard Drop*—which is to embody the following essential points:

a. Each drop test apparatus (Drop) must be standardized.

b. The ball (falling masses) shall weigh 1,000 or 1,500 pounds; the smaller is, however, preferable.

c. The ball may be made of cast iron, cast or wrought steel; the shape is to be such that its centre of gravity be as low as possible.

d. The striking block is to be made of forged steel, and is to be secured to the ball by dovetail and wedges in a rigid manner, and so that the striking face is placed strictly symmetrical about and normal to its vertical axis passing through the centre of gravity. Special permanent marks are to indicate the correctness of the face in these respects.

Special marks should be made to indicate the centre of the anvil block.

e. The length of guides on the ball should be more than twice the width between the guides, which are to be made of metal; *i.e.*, rails so placed that the ball has but a minimum amount of play between them.

Lubricating the guides with graphite is desirable.

f. The detachment or shears must not cause the ball to oscillate between the guides, and must be readily and freely controllable, with the point of suspension truly above the centre of gravity of the ball, and a short movable link, chain or rope, is to be fixed between the ball and shears or detachment.

The subjoined Fig. 69, showing the method officially employed in Russia, is particularly recommended.

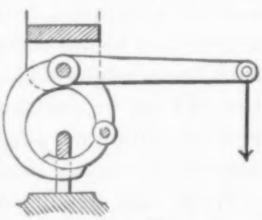

FIG. 69.

g. When a constant height of drop is used, an automatic detaching device is recommended.

h. The bearings for the test piece are to be rigidly attached to

the scaffold or frame, and they should be, wherever possible, in one piece with it.

i. The weight of frame, bearings, and anvil block should be at least ten times that of the ball.

k. The foundation should be inelastic, and consist of masonry, the magnitude of which is to be determined by the locality and subsoil.

l. The surface struck should always be accurately level; therefore proper shoes or bearing blocks are to be provided for testing rails, axles, tires, springs, etc., etc., to ensure a proper level upper surface; these blocks are to be as light as possible.

The exact shape of these bearing-blocks is to be given on each test report.

m. The gallows or frame should be truly vertical and the guides accurately parallel.

n. The height of fall of ball should be 20 feet clear, between striking and struck surfaces.

o. Drops which by friction of ball on guides absorb two per cent. of the work due to impact, are to be discarded.

p. For large tests a ball weighing 2,000 pounds is to be used.

q. A sliding scale is to be attached to the frame and in such a manner that the zero mark can always be placed on a level with the top of the test piece.

5. *Determination of those properties of a material which suggest its adoption.*

Materials are to be tested in such a manner as to develop a full knowledge of those qualities or properties which suggest their adoption for any particular purpose; and the tests are to embody, as closely as possible, actual conditions when in use.

The quality of a material itself is the resultant of all of its mechanical properties. As long as the interdependence of these several properties is unknown, so that the existence of several might lead to the deduction of others—and we are still far from such knowledge—so long the observation of a few of the properties will not be sufficient to judge of the applicability of the material for different purposes; it must therefore be tested under conditions exactly similar to those met with in actual use.

6. To the results recorded by the experimenter are to be annexed such short notes upon the machines and methods employed as are necessary, in judging of the results obtained.

7. The sources of test pieces, as well as microscopic or chemical

analyses, or both, as well as notes on the process of their manufacture and other physical, chemical, or technical characteristics, are to be noted on the Reports of Tests.

Such amplification of reports of tests will rarely be possible in routine testing, but it is highly desirable, and should never be neglected in scientific researches and standard tests.

8. Although the exact influence of duration of tests on results obtained has never been positively determined, enough data have been secured to show that time should be carefully noted in each case.

II. Tests of Wrought Iron and Steel.

A. Rails.

1. Rails—for reasons of safety of traffic, and in accordance with recommendations, and I. No. 5—should be subjected to the *drop test,* by means of proper devices (see Standard Drop, I. No. 4).

2. When further knowledge of the quality of the material is desired, *tension tests* are to be made.

3. Rails should be subjected to *bending tests* (with static load), and in two ways; one to determine the "*yield point,*" by permanent set; the other to determine the amount of permanent deflection under excessive loads beyond the elastic limit.

B. Axles.

1. Axles are to be subjected to drop test at their middle as well as the ends.

2. Tension Tests of same are to be made for additional knowledge of material.

3. Axles need not be subjected to bending test.

C. Tires.

1. Tires are to be subjected, the same as rails and axles, to the drop test.

2. Tension tests are to be made when additional information about the material is desirable.

3. The hammer test is not necessary.

D. Wrought Iron for Structural Purposes.

1. It should be subjected to tension test.
2. To the bending test.
3. Also to welding test.

E. Low Steels for Structural Purposes.

1. Tension tests and
2. Bending tests are to be made, as well as
3. Welding tests and
4. Hardening tests when the material is to be used for welded members, so commonly used in this country.
5. Annealing test of forged work.

F. High Steels for Structural Purposes.

1. Tension tests and
2. Bending test, as well as
3. Hardening tests, are to be made.

G. Wrought Iron for Boiler Work.

1. For three shapes used in boiler work of wrought iron the following tests are to be made:

a. For Plates.

1. Tension test.
2. Bending test; cold, hot.
3. Forging and punching test.

b. For Shape Iron.

1. Tension test.
2. Bending test; cold and hot.
3. Forging and punching test.
4. Welding tests for all shapes to be welded when in use.

For Rivet Rods.

1. Tension test.
2. Bending and forging test.

H. Low Steels for Boiler Work, are to be tested,

1. By tension test.
2. By the bending test; hot and cold.
3. Quenching or hardening tests.
4. Forging tests.

I. Materials used in Ship Construction.

1. *Plates*, are to be subjected

   a. To tension test.
   b. To bending test, cold.

2. *Shapes of all kinds* are to be tested
   a. By tension tests.
   b. By bending test; cold and hot.

3. *Rivet Iron* is to be subjected to
   a. Tension test.
   b. Bending test.
   c. Forging test.

J. Wire.

Wire is to be subjected to

1. Tension test.
2. Winding test by mechanical means, excluding arbitrary treatment.
3. Bending test, by *repeatedly bending* the wire forward and back around a turned stud having a diameter equal to the thickness of the wire to be tested.

K. Wire Rope.

Wire ropes are to be tested by

1. Tension test.
2. By impact longitudinally.

III. Cast Iron.

Cast Iron should be tested by

1. Tension test.
2. Bending test.
3. Compression test.

IV. Copper, Alloys, and Soft Metals.

Copper, its alloys and other soft metals, should be subjected to

1. Tension test.
2. Compression test.

V. Woods.

Woods are to be subjected to

1. Tension test.
2. Transverse test.
3. Compression test.

VIII. Method of Testing.

1. *General Recommendations.*

In every case where it is possible to test material in its finished form this should be done, as thus only are we often enabled to determine the value of technical processes to which material is subjected, and whether the material is of the proper kind to recommend its adoption. In order, however, to compare material generally, standard tests should always be made in addition to the above.

When material is always used in the condition in which it comes from the rolls or mould, it should be tested in such condition and in full-size pieces, and not by using specially prepared specimens. Thus the fact of cast metal having sometimes a decided skin may produce beneficial results, although in some cases the contrary is produced; therefore, in one case, chilling is resorted to and in the other, annealing. It will be seen that in either of these cases a specimen specially prepared would give erroneous results, and the only manner in which comparative results would be obtained would be to obviate all surface influences.

The same holds true in rolled material, and the method of procedure in both cases must depend largely upon the particular application of the material.

Therefore the method of procedure must begin with a study of the purposes for which the particular material is to be applied.

As, however, great quantities of material are used for similar purposes, and past experience has shown that there is a relation between tests on small specimens and pieces of large dimensions, it is generally sufficient to prepare standard specimens and test them by similar methods.

All samples submitted for test must be carefully examined for flaws, defects, and irregularities or abnormal appearances, and rejected for either these causes; when, however, no perfect material is available, then the peculiar characteristics are to be noted by sketch and description, if possible, before proceeding to prepare test specimen.

Preparation of test specimen can now be commenced in accordance with propositions relating thereto, as in the Appendices.

Careful and correct preparation of test pieces is so very important a matter that sufficient stress can hardly be laid upon it. The slightest blow of a hammer or simply dropping pieces on metal or stone are frequently causes of most erroneous results, and

often have greater effect on results obtained and quality of material than wide variations in composition or manipulation. As a general rule, however, all soft materials are less liable to show such effect than harder ones.

In preparation of wood specimens, it is of vital importance to avoid even the slighted nicks or cuts and bruises on surfaces and corners, as these will invariably be the causes of failure in all classes of tests.

All measurements in standard or scientific tests are to be made with the greatest possible care and highest degree of accuracy, assisted by the best apparatus.

2. *Tension Test.*

Knowing that the test piece is correctly prepared, the next process is to mark off the standard length and subdivisions, and then making careful note of all dimensions; recording total length, gauge length (or length on which measurements of strain are observed), fillets at shoulders, shape of shoulders, if any, length of part gripped, and polarity of material, if magnetic.

The specimen is to be divided between gauge marks into inches

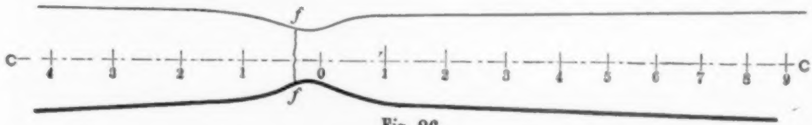

Fig. 26.

and half inches, in order that the stretch after rupture or elongation can be measured correctly. As material generally breaks at a point other than half-way between the gauge marks, direct measurements of stretch between them rarely give correct results or such which do justice to the material. As, however, the change of shape around the location of fracture is almost invariably symmetrical, allowance for eccentric fractures can be made. As is well known, the stretch of material near plane of rupture is very much greater than that at points distant from it; thus if rupture were to occur at a gauge mark, the total strain between marks would be very much less than when the rupture had taken place near the centre between them, as the stretch of material one inch each way from fracture is nearly double that of similar parts three or four inches distant. Therefore, a direct measurement of stretch would give an elongation manifestly unfair to the material, and only such tests

would be comparable in this respect in which rupture occurred in exactly similar locations.

If now the total stretch were determined by measuring the same in the manner following, a true elongation for the standard length between gauge marks would be obtained.

Thus, having divided the standard length into 24 *parts, measure from* ff *to cover all divisions up to* 12 *to the right—or twelve parts—then measure from* ff *toward the left to* 4, *and add the elongation of parts* 4–12, *measured to right of fracture, or eight parts to this; thus the elongation of the standard* 24 *parts will be obtained as though the fracture were located exactly at the middle division, for the elongation of the divisions* 4–12 *on the right is, undoubtedly, practically the same as that of a similar original length would be if located on the other side of the fracture.*

The direct measurement of elongation, as heretofore generally made, has always been unsatisfactory, and if given without statement of location of fracture, was very apt to be misleading; for this reason the above method is proposed and recommended, and that standard test pieces be divided into inches and one-half inches, as the most suitable for the purpose. It would give better results to divide the standard eight inch length into quarter inches or still smaller subdivisions, but as this would take more time and labor, the half inch divisions are recommended.

Having then determined the equilibrium of the testing machine before each test, the specimen is inserted in a truly axial position in the machine by measuring carefully its position in two directions in planes normal to each other and passing through the axis of the machine. This must be done with the greatest care to make sure that the stress will be applied symmetrically to the test piece.

Now the auxiliary apparatus for measuring stretch or obtaining autographic diagrams may be attached, and the actual test can be commenced.

The test is made by applying stress continuously and uniformly without intermission until the instant of rupture, only stopping at intervals long enough to make the desired observations of stretch and change of shape.

The stress should at no time be decreased and reapplied in a standard test, but should be maintained continuously.

After removing the fractured specimen from the machine, all measurements of shape are to be made, and a description of the peculiar characteristics of fractured as well as external surface be

given, adding the time of test. When fracture is cup shaped, state the position of cup—whether in upper or lower piece.

In recording the results of tests, loads at elastic limit, at yield point, maximum, and instant of rupture are all to be noted.

The load at elastic limit is to be that stress which produces a change in the rate of stretch.

The load at yield point is to be that stress under which the rate of stretch suddenly increases rapidly.

The maximum load is to be the highest load carried by the test piece.

The load at instant of rupture is not the maximum load carried, but a lesser load carried by the specimen at the instant of greatest stricture.

In giving results of tests it is not necessary to give the load per unit section of reduced area, because such figure is of no value; because it is not always possible to obtain the load at instant of rupture; because it is generally impossible to obtain a correct measurement of the area of section after rupture; and lastly, because the amount of reduction of area is principally dependent upon local and accidental conditions at the point of rupture. The modulus or coefficient of elasticity is to be deduced from measurements of strain observed between fixed increments of load per unit section; between 2,000 pounds per square inch and 12,000 pounds per square inch; or between 1,000 pounds per square inch and 11,000 pounds per square inch. With this precaution several sources of error are avoided and make it possible to compare results on the same basis.

By loading specimens up to 1,000 or 2,000 pounds per square inch, all initial errors are avoided, such as occur generally at the commencement of each test. The auxiliary apparatus adjusts itself somewhat during this period of loading and the specimen assumes a true position should any slight irregularity exist.

By measuring the stretch for an increment load of 10,000 pounds per square inch, a reasonable elongation is obtained and slight errors of observation are of less importance than when smaller increments of load are used, as the same error of observation would exist for measurements of strain due to increments of load of 2,000 and 10,000 pounds per square inch.

It would, in all cases of strong or hard materials in which the limit of elasticity is found above 30,000 pounds per square inch, perhaps be desirable to let this increment of load be 20,000 pounds

in order to measure a still greater stretch and thereby reduce possible errors; but as a number of materials have a lower elastic limit, the smaller increment of load is recommended in all cases.

There is, however, no objection to measure the stretch for the greater increment as well, though in that case this should be distinctly stated on the report.

Inasmuch as all materials are used within the elastic limit only, it cannot be urged with too much emphasis that the elastic properties within the limit should be observed with the greatest care as well as the ultimate behavior as heretofore mainly noted. All theories of the resistance of materials are based on the properties within such limit, and it is therefore recommended that they be investigated with proper apparatus in order to determine them with the greatest possible precision.

For some purposes, such, for instance, as working materials, their ultimate resistances should also be known, but generally the elastic resistances only are called into play in finished structures, except rubbing surfaces in machinery.

3. *Compression Tests.*

The test pieces are in all cases to be prepared with the greatest care to make sure that the end surfaces are true parallel planes normal to the axis of the specimen.

All materials are to be placed in the machines without the interposition of wedges, chains, or disks of materials of any kind and must be placed truly central or coincident with the axis of the machine.

In large specimens, when tested horizontally, initial flexure of test pieces is to be avoided by counter-weighting the mass of the pieces by proper devices. Although compression tests have heretofore been made on diminutive sample pieces, it is highly desirable that tests be also made on long pieces from 10–20 diameters in length, corresponding more nearly with actual practice, in order that elastic strain and change of shape may be determined by using proper measuring apparatus.

The elastic limit, modulus or coefficient of elasticity, maximum and ultimate resistances, are again to be determined, as well as the increase of section at various points, viz.: at bearing surfaces and at crippling point.

The total compression is to be determined by dividing the long specimens precisely as in the case of tension test pieces, and the

modulus or coefficient of elasticity in the same manner as before described.

The use of long compression test pieces is recommended, because the investigation of short cubes or cylinders has led to no direct application of the constants obtained by their use in computation of actual structures, which have always been and are now designed according to empirical formulæ obtained from a few tests of long columns.

Moreover, the investigations of Christie, and also of C. A. Marshall, have shown a direct relation between the elastic properties and resistances of materials when tested in long specimens.

4. *Transverse Test.*

In this test the specimen is to be placed in the machine in a truly horizontal manner; stress is to be applied at the centre normal to the axis of test piece, and in a plane passing through the three points of resistance. Sharp edges on all bearing pieces are to be avoided, and the particular form selected to be shown by sketch or full description on report.

The use of rolling bearings which move accurately with the angular deflection of ends of bars are especially recommended, as fixed bearings cause change of length of specimens between points of support during test.

Deflections at centre are to be measured carefully from a base or table which does not change its position relative to bearing surfaces under varying loads. It is also desirable to observe deflections at fixed distances from centre for the purpose of plotting the elastic curve of the material within the elastic limit.

It is of primary importance to measure deflections within elastic limit, although a knowledge of permanent deflections is sometimes valuable.

5. *Torsion Tests.*

In this test particular care must be had that the body and ends of test piece are absolutely concentric; that the stress be absolutely axial and applied tangentially to the surface of test piece, and that no displacement in this respect occur during test. There must be free motion of the ends along the axis so as to avoid longitudinal stress on the material.

The angular motion must be carefully noted by the use of indicators placed on the body of the test piece and not on the shoulders.

Particular attention must be given to distortion within elastic limit.

6. *Multiple or Piece Test.*

At present, no general recommendation can be offered how to apply this test except as referring to small rods and bars up to about one square inch section, and finished articles such as springs, chains, pipes, boilers, and cylinders of every kind. In case of *small bars and rods*, a satisfactory and inexpensive test that can be applied rapidly is as follows: One end of the bar is inserted into a bending test machine, shown by Fig. 27. This machine carries a central

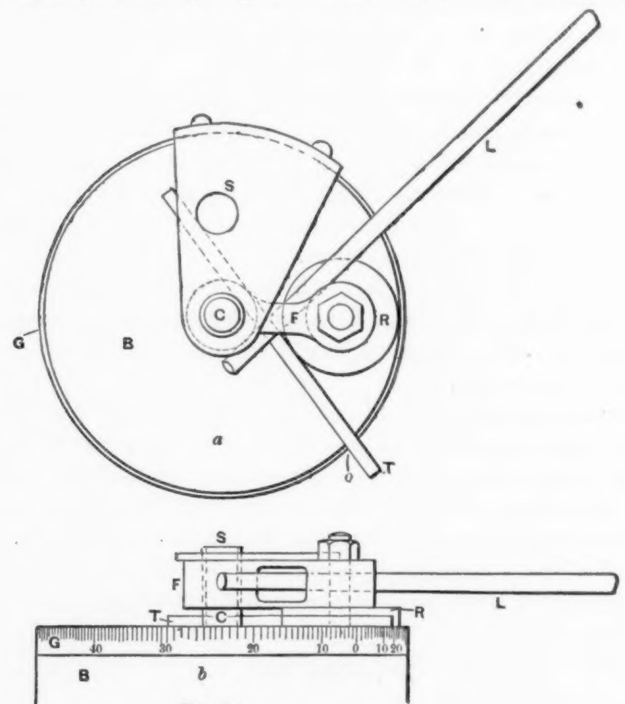

Fig. 27.

stud of adjustable size, around which the bar or rod is bent by a steady pressure and up to a given angle adopted, corresponding to the relative quality of the particular material examined.

The adjustable stud *C*, should have a diameter equal to twice the thickness or diameter of the material to be tested. The material is placed as shown at *S T;* then roller, *R*, carried by frame, *F*, is brought to bear against material by means of lever, *L*, passing

through frame, F. Original position of bar is read off on graduated circle, G, which can revolve so as to bring O opposite material if desired. Now pressure on lever, L, bends material, T, until the desired permanent set, as read off on G, is produced.

This test is very rapid, wastes a minimum amount of material, and requires an inexpensive machine, which is also used for bending tests in general.

Springs should be tested by applying the maximum working load, measuring length or height of spring, before and after, as well as dimensions (extension or compression) during the test. According to requirements to be fulfilled, the pressure is to be produced by impact or by steady pressure. A single application of test load suffices.

Chains are to be tested by subjecting them to a working load and observing elastic and permanent strain and change of shape of links.

Pipes are to be tested by applying hydraulic pressure equal to a maximum working load, observing expansion of pipe under such load, as well as permanent set if any, and noting that pipe does not leak but carries pressure.

Cylinders are to be tested in a similar manner to pipes, and *boilers* according to the method adopted by the Hartford Steam Boiler and Inspection Co., and now adopted generally throughout the country.

Copies of instructions giving an account of this method are so widely distributed that it is not considered necessary to append them in this place.

7. *Welding Test.*

Although not often applied in Europe, this becomes of importance in our country because welded members are so generally used in structural work. The test is made as follows: A piece of the metal about one inch in largest dimension of section is prepared as for a simple scarf weld. The two halves to be welded are then inserted in a coke, gas, or petroleum furnace having a true reducing flame, having ascertained that the furnace and fire are perfectly clean. When the metal has attained a clear white or welding heat, and without further delay, it is to be removed from the fire, and then by a few blows of a hammer weighing from 8 to 10 lbs. to be welded together to form a joint. Then several blows delivered to the end of the bar will serve to upset it, and the weld must then be drawn down to the normal size

and shape by the use of the same hammer. The use of a small hammer is not admissible, as with it the best results cannot be obtained. The whole operation of welding and finishing is to be done in one heat without the use of fluxes of any kind. When the bar has become cold without the use of water, it is to be tested in its rough shape by the regular tension test. Another sample is to be welded in the same manner, and after nicking it to the depth of the weld, it is to be bent or broken, if possible, to show the character of the welded surfaces. No water is to be used at any time during the test and the bar must neither be finished cold nor struck by the hammer except where well heated.

8. *Bending Tests*

Afford the readiest means of determining ductility of materials, and are valuable when made by mechanical means.

This test is to be made as described under 6, p. 626, by means of an apparatus similar to the one shown in Fig. 27. The apparatus must work free from shock and injury to the bar by coming into contact with sharp corners or edges. The old method of making bending tests by sticking the test piece into a hole in the anvil or the jaws of a vise, and then bending the free end by striking it by a hammer is to be deprecated most seriously.

The stud about which the material is to be bent is to measure twice the thickness or diameter of the bar, and the diameter of stud, Fig. 27, is to be varied by using sleeves which fit the stud closely. The amount of permanent deflection varies according to the quality of the material and its uses, and is read off directly on the graduated ring, G.

This apparatus avoids possible injury to the material, makes application of stress uniform, and gives correct measurements of diameter around an angle to which deformation is produced, besides providing a rapid method easily carried out.

9. *Hardening Tests*

Are to be made by making the foregoing bending tests, only carrying it on to rupture, and reading the angle at which rupture occurs; then other pieces exactly similar to those thus tested are to be heated carefully to a fair red heat and then plunged into water at a temperature of from 32–40° F. These quenched pieces are then to be tested precisely as those tested in their natural condition, and the angle through which they bent before

fracture occurred is to be carefully noted; the difference in the amount of possible flexure in the two cases represents the amount of hardening produced.

10. *Forging Test.*

This test is advisable to be used on rivet rods principally, and is to be made as follows: A part of a rod is brought to a fair red heat and then hammered by an ordinary hammer until cracks barely begin to show at the edges of the material. The relative amount of flattening before cracks appear indicates the red shortness of the material.

11. *Punching Tests*

Are made by measuring the width of metal between the finished—not rough sheared—edge to the near edge of a punched hole which is necessary to prevent splitting of the material, measured on the punch side.

12. *Abrasion Tests.*

No recommendations can as yet be made as to possible methods for abrasion tests of rails or other materials subject to surface wear.

IX. Standard Shapes of Test Pieces.

1. *Standard Shapes for Tension Test Piece.*

Inasmuch as a standard shape, similar in all essential particulars to that in general use in the United States, has been adopted by the conferences heretofore mentioned, it is recommended that the same shape be adopted here, for all scientific or standard tests. For routine or practical work specimens having similar general dimensions, diameter and gauged length, are to be used; giving results sufficiently comparable for general purposes. Specimens for scientific or standard tests are to be prepared with the greatest care and accuracy, and turned according to the following dimensions, intended to represent metrical dimensions as nearly as possible, eight inches being 200 mm. for all practical purposes in this respect.

The specimens are to be held by true bearing on the end shoulders, as all gripping or holding devices in common use produce undesirable effects on the cylindrical part. These test pieces are to have different diameters, according to original thickness of material, and to be exactly 0.4, 0.6, 0.8, and 1.0 in. diameter, but

for all these different diameters the angle and length of coned neck is to remain as shown in Fig. 28, diameter of larger end and head to change directly in proportion to change of diameter. For routine work the gauged length is to be as above, invariably 8 in.; the cylindrical part is to run into the shoulders by a conical part rather than a fillet, as is now commonly the case.

This form is that recommended for all metals except copper and its alloys, for which no standard shape can as yet be suggested, as experiments heretofore made have produced no definite results in this respect.

When rough bars—as material comes from the rolls—are to be

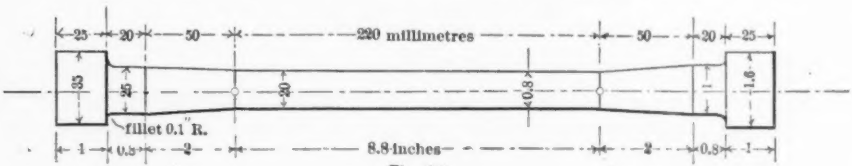

Fig. 28.

tested, the standard gauged length of 8 in. is to be invariably used for measurements of strain, as heretofore explained, and the length of specimen from gauge mark to nearest holder or shackle is to be not less than one diameter of material in round bars; not less than one and one-half times the width for flat specimens cut in strips with parallel sides, and not less than one and one-half times the width of side of square bars.

For flat specimens cut from larger sections a milled shape, as shown below, Fig. 29, is the most appropriate.

Shoulders to be ample for obtaining a full grip of shackles or holders.

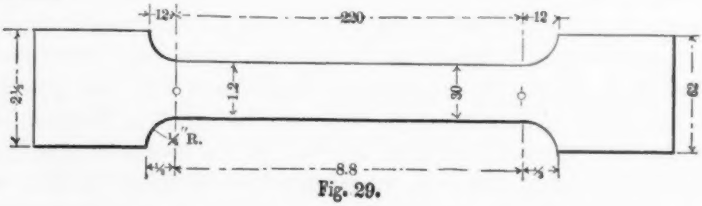

Fig. 29.

When specimens are cut from plates and shapes in which the thickness is fixed, the width is to remain 1.2″ up to a maximum thickness less than one inch. When thickness is one inch or more, then the specimen is to be cut so as to have a width of 0.4 in. and a thickness equal to original thickness of material. When testing

plates and shapes the roll surface is always to be retained, and duplicate specimens cut from shapes are to be so selected that they represent their entire width.

2. *Compression Test Specimens.*

Specimens for compression tests are to be cylinders two inches in length and one inch in diameter, when ultimate resistance alone is to be determined. For all other purposes, especially when elastic resistances are to be ascertained, specimens of one inch diameter and ten and twenty inches length are to be used. Standard length on which strain is to be measured is to be 8 in., as in tension tests. In all cases, the greatest care is to be used in squaring the ends and having a perfect surface.

3. *Transverse Test Specimens.*

For transverse tests, bars one inch square and forty inches long are to be used, the bearing blocks or supports to be exactly thirty-six inches apart—centre to centre. For transverse tests of wood, sticks 3 inches wide and deep, being exactly square and carefully planed, 40 inches long, 36 inches centre to centre of bearings, are to be used.

For standard or scientific tests of cast iron, such bars are to be cut out of a casting, at least two inches square, or two and one-quarter inches in diameter, so as to remove all chilling effect. For routine tests cast bars one inch square may be used, but all possible precautions must be taken to prevent surface chilling and porosity.

4. *Torsion Tests.*

For torsion tests the cylindrical specimen with square shoulders is to be avoided, as leading to constant error.

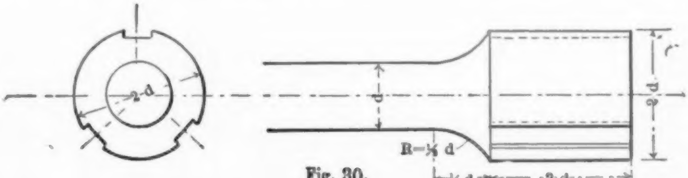

Fig. 30.

A cylindrical specimen, with cylindrical concentric shoulders, the two connected by large fillets, is to be used; see Fig. 30.

This specimen is to be held by keys and keyways as shown, not more than $\frac{1}{8}$ in. deep, cut in shoulder, which has a diameter and

length equal to twice diameter of specimen up to one inch; distance from shoulder to gauge marks is to be not less than one diameter. A standard length between gauge marks is as yet undetermined and cannot as yet be recommended.

5. *Bending Tests.*

Specimens for bending tests are to be flat strips with parallel sides and one inch wide. In case of small rods, round or square, not over one inch, rough pieces as the material comes from the rolls are sufficient.

6. *Multiple, Welding, and Abrasion Tests.*

For these no standard shapes of test piece can be recommended. Generally a piece of original shape is sufficient.

All of which is respectfully submitted.

GUS. C. HENNING,
Reporter to the Committee.

ADDENDUM I. TABLE I.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY................................, OF................................

Testing Machine built by............... When procured,.......... Maximum capacity,........lbs. How driven,...............

Position of specimen (horizontal or vertical),......................

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			DIMENSIONS BEFORE TEST.			LENGTH BETWEEN GAUGE MARKS.						STRESS IN LBS.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	REMARKS.

NOTE.—All dimensions to be expressed in inches and decimals of *inches*.
The Elastic Limit to be determined by noting the point at which the *rate* of elongation *suddenly* increases. All increment of load should then be stopped, and careful observations made of the length, minimum section, and stress. After noting these, the test can be resumed and continued to rupture.

(*Signature*).................................

....

.................................1886.

ADDENDUM I. TABLE II. *a*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY C. A. MARSHALL, IN CHARGE OF TESTING LABORATORY OF CAMBRIA IRON CO., JOHNSTOWN, PA.

Testing machine built by J. L. Gill, Jr. When procured, about 1880. Maximum capacity, 100,000 lbs. How driven, gearing drives nut on screw of lower head. Position of specimen (horizontal or vertical), vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			Dimensions before Test.			Length between Gauge Marks.						Stress in Lbs.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	Remarks.
Iron, 2394	....	....	″ ..	″	″	″	″	″	......	″	....	......	......	......	....	Tested on Emery 300,000 lb. machine.
Iron, 2395	Cold Rolled Iron.	¾″ Rd. Straight.	17	.748 diam.	9	8.00	8.017425	8.87	......	.555 diam.	¼″ from 8″ mark.	25960	34035	......	52 min.	Fracture, silky, fibrous. See detail sheet.
Iron, 2396	Cold Rolled Iron.	¾″ Rd. Straight.	17	.748 diam.	9	8.00	8.017300	8.54	......	....	Outside marks.	25960	34580	28000	30 min.	Fracture, silky, fibrous. See detail sheet.
Steel, 2397	Cold Rolled Steel.	¾″ Rd. Straight.	17	.749 } mean .748 } .7485	9	8.01	8.026600	9.02	......	.488 diam.	4″ from 8″ mark.	25080	32800	25000	29 min.	Fracture, ¾ cup. Fine silky. See detail sheet.
Steel, 2398	Cold Rolled Steel.	¾″ Rd. Straight.	17	.752 } .748 } = .750	9	8.01	8.028275	8.50	......	.531 diam.	⅛″ outside 8″ mark.	28288	35600	28000	34 min.	Silky edges, dull, with specks crystalline. See detail sheet.
Steel, 2399	Cold Rolled Steel.	¾″ Rd. Straight.	17	.749 } .747 } = .748	9	8.00	8.019600	8.58	......	.478 diam.	Exactly at 8″ mark.	30800	36250	25200	30 min.	Full cup, fine silky. See detail sheet.

ADDENDUM I. TABLE II. *b*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY C. A. MARSHALL, IN CHARGE OF TESTING LABORATORY OF CAMBRIA IRON CO., JOHNSTOWN, PA.

Testing machine built by Yale & Towne Mfg. Co. When procured, 1885. Maximum capacity, 300,000 lbs. How driven, hydraulic power, from adjustable stroke, 3-plunger pump. Position of specimen (horizontal or vertical), vertical.

2403	Steel.	2½″ × ½″ Straight.	22	2.520 × {.508, .502, .502} Mean, .504	12	8.005	8.015	10.59	......	1.837 × {.245, .313, .345} Mean, .329	3¼″ from 8″ mark.	47300	69100	60000	....	Cup, silky fracture. Reduction of area = 52.4%.
2404	Steel.	2½″ × ½″ Straight.	22	2.520 × {.501, .500, .506} Mean, .502	12	8.01	8.02	10.545	.....	1.885 × {.352, .303, .370} Mean, .332	4½″ from 8″ mark.	47900	74500	64400	....	Fracture, ½ cup, ½ 45°, silky. Reduction of area = 50.5%.
2405	Steel.	2½″ × ½″ Straight.	22	2.520 × {.505, .495, .500} Mean, .500	12	8.02	8.03	10.66	......	1.897 × {.333, .356, .356} Mean, .344	5½″ from 8″ mark.	46200	67400	64000	5¼ min. to limit. 7 min. to breaking 12¼ min., total.	Fracture, 45°, silky. Reduction of area = 48.2%.
2400	Steel.	1″ Sq. Straight.	22	.990″ × .990″	10¼	8.01	8.018850	9.46	......	.790 × .780	4½″ from 8″ mark.	30380	49100	30000	12 min. to limit. 13 min. to rupture. 25 min., total.	Fracture, silky edges, mushy interior. See detail sheet.
2401	Steel.	1″ Sq. Straight.	22	.990″ × .990″	10¼	8.00	8.008450	10.52	......	.685 × .685	4½″ from 8″ mark.	30870	51300	42000	26 min.	Fracture, 45°, silky. See detail sheet.
2402	Steel.	1″ Sq. Straight.	22	1.011 × .973	10¼	8.00	8.008950	10.55	......	.690 × .655	3″ from 8″ mark.	33458	51600	42000	48 min.	Fracture, ½ cup, silky. See detail sheet.
Iron, 2394	Cold Rolled Iron.	¾″ Rd. Straight.	17	.748″ diam.	9	8.00	8.016875	8.69	......	.530″ diam.	1½″ outside of 8″ mark.	25960	*34250	......	56 min.	Fracture, silky, fibrous. Broke in grip. See detail sheet.

* This taken on Gill Testing Machine.

NOTE.—All Dimensions to be expressed in inches and decimals of *inches*.

The Elastic Limit to be determined by noting the point at which the *rate* of elongation *suddenly* increases. All increment of load should then be stopped, and careful observations made of the Length, Minimum Section, and Stress. After noting these, the test can be resumed and continued to rupture.

November 18, 1886.

C. A. MARSHALL.

ADDENDUM I. TABLE II. *c*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY THE ORDNANCE DEPT., U. S. A., OF WATERTOWN ARSENAL, MASS.

Testing machine built by A. H. Emery. When procured, 1879. Maximum capacity, 800,000 lbs. How driven, hydraulic power. Position of specimen (horizontal or vertical), horizontal.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			DIMENSIONS BEFORE TEST.			LENGTH BETWEEN GAUGE MARKS.						STRESS IN LBS.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	REMARKS.
			"	" "	"	"		"		" "	"					With specimens of the cross-section dimensions herein reported, a minimum length of 36" is required to obtain all the data provided for on this blank form, unless the specimens are provided with threaded ends, in which case the length of stem may be from 2" upward. It is necessary to have finished surfaces in order to determine data for column 10.
967	Steel..	Round.	17	.722 diam.	10	8	......	10.27	......	....	2.90	Specimens too short.	33,200	27.400	4'	
968	Steel..	Round.	17	.724 diam.	8	8	......	8.59	......	.48 + diam.	3.90		35,490	27.600	4¼'	
969	Steel..	Round.	17	.724 diam.	8	8	......	8.32	......	.50 + diam.	3.80		34,540	25,500	4'	
970	Iron ..	Rcund.	17	.721 diam.	8	8	......	8.31	......	.53 + diam.	3.80		34,470	26,400	4¼'	
971	Iron ..	Round.	17	.723 diam.	8	8	......	8.84	......	.56 + diam.	6.80		35,720	27,200	3¾'	
972	Iron ..	Round.	17	.723 diam.	8	8	......	8.29	......	.52 + diam.	3.00		33,350	24.900	3½'	
973	Steel..	Flat ...	22	2.51 × .52	10	8	......	10.66	......	1.89 × .36	12.00	48,200	75,080	64.000	5'	
974	Steel..	Flat ...	22	2.52 × .50	10	8	......	10.63	......	1.92 × .36	10.80	46,500	67.960	61,000	4'	
975	Steel..	Flat ...	22	2.52 × .52	10	8	......	10.39	......	2.01 × .40	11.60	47.700	74,[illegible]20	70,000	4½'	
976	Steel..	Square	22.1	.99 × .99	10	8	......	9.68	......	.77 × .84	12.30	33,200	50,950	48.000	4½'	
977	Steel..	Square	21.9	.98 × .99	10	8	......	10.62	......	.67 × .68	11.30	30,780	52,890	46,300	5'	
978	Steel..	Square	22.7	.99 × .99	10	8	......	10.68	......	.64 × .65	11.20	32,350	51,410	40.000	4¼'	

November 6, 1886.

F. H. PARKER,
Major Ordnance Dept., U. S. A., Commanding.

ADDENDUM I. TABLE II. *d*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY CHAS. H. MORGAN, OF THE WASHBURN & MOEN MFG. CO., WORCESTER, MASS.

Testing machine built by Alb. H. Emery, Yale & Towne Mfg. Co. When procured, 1885. Maximum capacity, 70,000 lbs. How driven, by hydraulic pressure. Position of specimen (horizontal or vertical), horizontal.

1	Steel..	Round.	17	.750	8.53	8	8.0269	8.84	.7483	.490	7.3	33,500	38,000	36,000	7 minutes.	
2	Steel..	Round.	17	.750	8.78	8	8.0265	9.03	.748	.520	6.9	28,000	33,000	32,100	6 minutes.	
3	Steel..	Round.	17	.7495	8.58	8	8.0196	9.15	.747	.500	7.4	26,000	32,000	31,600	6 minutes.	
4	Iron ..	Round.	17	.749	8.98	8	8.475	8.56	.746	.550	3.5*	29,000	35,000	34,200	8 minutes.	Flaw 0.5 inch from the rupture.
5	Iron ..	Round.	17	.749	8.48	8	8.0290	8.99	.746	.550	9.2	28,000	34,200	33,000	6 minutes.	* Broken between the jaw and point of fracture.
6	Iron ..	Round.	17	.7485	8.58	8	8.0292	9.09	.747	.560	9.3	28,000	34,400	32,600	6 minutes.	

October 27, 1886.

KARL JANSSON,
Tester.

ADDENDUM I. TABLE II. *e*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY THE YALE & TOWNE MANUFACTURING CO., OF STAMFORD, CONN.

Testing machine built by the Yale & Towne Manufacturing Co. When procured, 1884. Maximum capacity, 151,000 lbs. How driven, hydraulic. Position of specimen (horizontal or vertical), vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
No. or mark of specimen.	Kind of material.	Form of specimen.	DIMENSIONS BEFORE TEST. Total length.	DIMENSIONS BEFORE TEST. Section.	Length between grips or jaws when starting test.	LENGTH BETWEEN GAUGE MARKS. When starting test.	LENGTH BETWEEN GAUGE MARKS. At elastic limit while under strain.	LENGTH BETWEEN GAUGE MARKS. After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	STRESS IN LBS. At elastic limit.	STRESS IN LBS. Maximum observed.	STRESS IN LBS. At time of rupture.	Duration of test.	REMARKS.
			″	″ ″	″	″		″		″ ″	″				Min.	
A	Steel...	Square...	22	1.014 × .980	12	8.010	....	10.530	......	.700 × .700	12.750	34,780	51,970	41,340	20	
B	Steel...	Square...	22	1.000 × .992	12	8.003	....	10.130	......	.710 × .710	7.750	33,290	52,240	43,990	18	
C	Steel...	Square...	22	.995 × .996	12	8.009	....	10.650	......	.675 × .675	13,250	32,900	51,060	41,000	14	
D	Iron....	Round ...	22	.75 diam.	12	8.000	....	8.680	......	.560 diam.	11.000	10,580	35,800	29,100	18	
E	Iron....	Round ...	22	.75 diam.	12	8.000	....	....	......	....	....	10,930	34,700	......	19	Broke in holder.
F	Iron....	Round ...	22	.75 diam.	12	8.000	....	....	......	....	....	11,000	35,300	30,100	12	Broke in holder.
G	Steel...	Round ...	22	.75 diam.	12	8.000	....	8.790	......	.480 diam.	9.250	15,270	37,800	25,200	10	
H	Steel...	Round ...	22	.75 diam.	12	8.000	....	8.900	......	.480 diam.	9.000	16,100	36,600	27,000	12	
I	Steel...	Round ...	22	.75 diam.	12	8.000	....	....	......	....	....	16,000	30,400	27,000	14	Broke in holder.
J	Steel...	Flat......	22	2.500 × .500	12	8.002	....	10.400	......	.350 × 1.900	11.700	47,540	75,220	68,200	25	
K	Steel...	Flat......	22	2.500 × .500	12	8.006	....	10.400	......	.330 × 1.870	12.350	48,700	75,580	65,600	15	
L	Steel...	Flat......	22	2.500 × .500	12	8.001	....	10.400	......	.340 × 1.900	12.150	48,690	75,340	67,600	18	

January 5, 1887.

G. L. BACKSTROM,
Operator.

YALE & TOWNE MANUFACTURING CO., HENRY R. TOWNE, *Pres*.

ADDENDUM I. TABLE II. *f*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY POTTSVILLE IRON & STEEL CO., OF POTTSVILLE, PA.

Testing machine built by Messrs. Riehlé Bros. When procured, ... Maximum capacity, 50,000 lbs. How driven, by hand hydraulic machine, single-throw pump. Position of specimen (horizontal or vertical), vertical.

1	Iron ...	¾″ Round	......	$.747_0$	9	8	....	9.93	......	$.543_0$	4½	72,300	81,990	......	8	Fibrous.
2	Iron ...	¾″ Round	......	$.747_0$	9	8	....	8.99	......	$.536_0$	4	63,880	91,100	......	5	Fibrous.
3	Iron ...	¾″ Round	......	$.746_0$	9	8	....	8.96	......	$.553_0$	8	67,260	83,820	......	4	Defective fibrous.
4	Steel...	¾″ Round	......	$.749_0$	9	8	....	9.06	......	$.492_0$	7½	66,090	88,290	......	4½	Silky.
5	Steel...	¾″ Round	......	$.743_0$	9	8	....	8.97	......	$.492_c$	6	62,470	73,800	......	4	Silky.
6	Steel...	¾″ Round	......	$.745_0$	9	8	....	8.92	......	$.496_0$	9	71,820	83,290	......	4½	Silky.

May 5, 1887.

WM. R. WEBSTER,
Assistant General Manager.

ADDENDUM I TABLE II. *g.*

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY CHESTER ROLLING MILLS, OF THURLOW, DELAWARE CO., PA.

Testing machine built by Riehlé Bros., Philadelphia. When procured, 1883. Maximum capacity, 102,000 lbs. How driven, hand-power gearing, hydraulic cylinder, and 3 pumps. Position of specimen, vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			Dimensions before Test.			Length between Gauge Marks.						Stress in Lbs.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	Remarks.
			"		"	"					"				Min	
1	Iron ..	¾ Round..	17	.748 diam. = .4394	10 3/16	8	.02	8.96	.4362	.2273	8½ top.	17,800	34,250	34,000	13	Elastic limit probably too low.
2	Iron ..	¾ Round..	17	.748 diam. = .4394	10 3/16	8	.02	9.017	.4382	.2367	8¼ top.	22,000	35,760	29,000	12	Elastic limit probably too low.
3	Iron ..	¾ Round..	17	.748 diam. = .4394	10 3/16	8	.05	9.017	.4359	.2349	8½ bot.	28,500	35,320	28,600	8	
4	Steel..	¾ Round..	17	.750 diam. = .4417	10 3/16	8	.04	8.575	.4359	.1924	4¼ bot.	28,500	38,000	30,800	9	Broke at lower gauge mark.
5	Steel..	¾ Round..	17	.748 diam. = .4394	10 3/16	8	.05	9.048	.4347	.1720	8 bot.	27,000	30,600	22,100	12	
6	Steel..	¾ Round..	17	.748 diam. = .4394	10¼	8	.04	8.404	.4359	.1955	3 top.	28,000	32,500	22,950	12	Broke above upper gauge mark.
7	Steel..	1 Square.	18	1.015 × .979 = .9937	9¾	8	.04	10.548	.9917	.4312	8¾ top.	34,400	52,850	44,000	18	
8	Steel..	1 Square.	18	1.013 × .977 = .9897	9¾	8	.04	10.62	.9869	.4121	9¼ bot.	32,500	52,100	44,000	18	
9	Steel..	1 Square.	18	.992 × .989 = .9811	9¾	8	.10	9.256	.9702	.6272	10¼ bot.	31,500	48,230	48,230	7	
10	Steel..	2½ × ½ Flat	18	1.017 × .519 = .5278	9	8	.06	10.285	....	.2119	8¼ top.	19,160	31,210	25,500	11	
11	Steel..	2½ × ½ Flat	18	1.02 × .5 = .51	8¾	8	.055	10.058	....	.2278	8½ top.	19,610	31,100	25,800	17½	
12	Steel..	2½ × ½ Flat	18	1.023 × .504 = .5156	9	8	.06	10.117	....	.2242	9¾ top.	19,690	31,330	25,830	12	

November 1, 1886.

LOWDON W. RICHARDS,
Inspector.

ADDENDUM I. TABLE II. *h*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY RIEHLÉ BROS., OF PHILADELPHIA, PA.

Testing machines built by Riehlé Bros. When procured, ... Maximum capacity, 120,000 lbs., one of 50,000 lbs. How driven, by hand, hyd. pump, and screw power machine (steam power). Position of specimen (horizontal or vertical), both vertical machines.

								Outside		Diam.						
1	Steel..	Round....	17	.748=.4394	13½	8	8.015	18.	.746	.47	4	27,000	30,750	21,540	30	Broke outside witness marks, but not in grips.
2	Steel..	Round....	17	.746=.4379	9½	8	8.015	9.16	.746	.48	8¼	30,000	33,700	24,200	18	
3	Iron ..	Round....	17	.749=.4400	9½	8	8.015	8.70	.748	.54	5	27,800	35,000	28,370	16	
4	Iron ..	Round....	17	.749=.4400	9½	8	8.015	8.80	.748	.53	6¾	27,600	33,700	26,480	20	On 50,000 lb. hand, double acting hyd. pump. T-machine.
5	Iron ..	Round....	17	.749=.4400	9½	8	8.015	9.	.748	.54	7½	27,500	34,300	27,900	16	
6	Steel..	Round....	17	.749=.4400	9½	8	8.015	9.30	.748	.47	3¾	28,500	33,300	24,080	18	
1	Steel..	Flat	22	.50 × 2.51	10½	8	8.015	10.54	2.51 × 50	1.90 × 33	10¾	47,300	75,600	67,600	25	
2	Steel.	Flat	22	.51 × 2.51	10½	10	10.015	12.76	2.51 × 50	1.98 × 37	8½	48,500	75,200	71,140	26	
3	Steel..	Flat	22	.52 × 2.51	10½	8	8.015	10.57	2.51 × 52	1.82 × 31	9½	50,500	75,300	64,400	24	120,000 lb. screw power T-machine driven by belt.
1	Steel..	Square...	22	.99 × 98	10¼	8	8.015	10.60	.98 × 99	.67 × 68	10¾	32,000	51,400	42,800	20	
2	Steel..	Square...	22	.99 × 99	10½	8	8.015	10.87	.98 × 99	.63 × 61	13½	31,800	51,200	40,000	20	
3	Steel..	Square...	22	.99 × 98	10½	8	8.015	10.68	.99 × 98	.63 × 63	12½	31,500	51,100	40,800	22	

December, 1886.

H. B. RIEHLÉ.
C. E. BUZBY,
Superintendent.

ADDENDUM I. TABLE II. ı.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY SIBLEY COLLEGE, CORNELL UNIVERSITY, OF ITHACA, N. Y.

Testing machine built by Fairbanks. When procured, September, 1886. Maximum capacity, 50,000 lbs. How driven, by belt. Position of specimen (horizontal or vertical), vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			Dimensions before Test.			Length between Gauge Marks.						Stress in Lbs.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	Remarks.
No.				Sq. Inches.	"				Sq. Inches.						Hours.	
1	Steel...	Round.....	17.03	.3685	10.5	8	8.0181	8.9	.3678	.1712	8.8	23,400	28,450	21,090	2	
2	Iron....	Round.....	17.01	.3696	10.62	8	8.0174	8.84	.3691	.1978	8.75	21,100	28,420	22,170	1.5	
3	Steel...	Round.....	17.	.3693	10.25	8	8.0171	8.86	.3691	.1548	5.12	24,000	28,320	19,790	1.5	
4	Iron....	Round.....	17.03	.3691	10.59	8	8.0153	8.83	.3685	.1885	8.84	20,200	27,860	21,690	1.5	
5	Steel...	Round.....	22.05	.4844	15.37	8	8.0096	9.94	.4839	.2419	11.95	16,000	26,210	21,530	1	
6	Steel...	Round.....	21.72	.4844	11.84	8	8.0090	9.84	.4839	.2669	11.75	16,000	25,380	21,470	1	
7	Steel..	Round.....	21.87	.4844	12.02	8	8.0084	9.86	.4839	.2341	11.95	16,000	26,430	21,000	1	
8	Steel...	Flat........	21.92	.4994	14.	8	8.0105	9.79	.4962	.2907	11.85	19,000	28,350	26,140	1	
9	Steel...	Flat........	21.98	.4991	14.3	8	8.0113	9.74	.4980	.2830	11.85	19,300	28,830	23,400	1	

R. H. THURSTON
Director.

ADDENDUM I. TABLE II. *k.*

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY EDGE MOOR IRON CO., OF EDGE MOOR, DELAWARE.

Testing machine built by Fairbanks. When procured, February, 1883. Maximum capacity, 100,000 lbs. How driven, belt. Position of specimen, vertical. Squares and rounds held by grooved wedges. Flats held by flat wedges.

			"		"	"					"				Min.	
1	Steel...	2" × ½" Flat	16	.516 × 1.99 = 1.02	9¼	8	8.05	10.10	.51 × 1.985 = 1.01	.365 × 1.60 = .584	8¼	38,900	60,500	60,500	15	Elongation in all tests measured with dividers and steel standard graduated into 100ths of an inch.
2	Steel...	2" × ½" Flat	16	.50 × 1.99 = .995	9¼	8	8.03	10.30	.49 × 1.98 = .97	.34 × 1.45 = .493	9½	38,200	55,700	55,700	15	
3	Steel...	2" × ½" Flat	16	.495 × 1.99 = .985	9¼	8	8.06	10.25	.48 × 1.98 = .95	.33 × 1.47 = .485	8¾	37,900	54,850	54,850	15	
4	Steel...	1" Square..	16	.99 × .99 = .98	8¾	8	8.05	10.43	.975 × .975 = .95	.64 × .715 = .46	9	33,90.	52,400	52,400	20	
5	Steel...	1" Square..	16	.99 × .99 = .98	8	8	8.02	10.40	.985 × .985 = .97	.66 × .63 = .41	9½	33,100	50,900	50,900	15	
6	Steel...	1" Square..	16	.995 × .995 = .99	8	8	8.07	10.60	.98 × .98 = .96	.62 × .63 = .39	9	33,800	51,400	51,400	15	
7	Iron .	¾" Round..	17	.745 = .4359	8	8	8.07	9.00	.735 = .4242	.55 = .2375	8¾	20,500	34,400	34,400	10	Selected before testing for iron.
8	Iron ...	¾" Round..	17	.745 = .4359	8	8	8.05	8.99	.74 = .4300	.575 = .2596	8	18,900	35,500	35,500	8	
9	Iron ...	¾" Round..	17	.475 = .4359	8	8	8.01	8.97	.74 = .4300	.57 = .2551	7	19,600	34,500	34,500	7	
10	Steel...	¾" Round..	17	.475 = .4359	8	8	8.06	8.83	.73 = .4185	.53 = .2206	5¼	20,400	34,300	34,300	8	
11	Steel...	¾" Round..	17	.475 = .4359	8	3	8.04	8.65	.735 = .4242	.49 = .1875	6	22,500	37,500	37,500	6	
12	Steel...	¾" Round..	17	.475 = .4359	8	8	8.03	8.85	.738 = .4277	.48 = .1809	8¼	23,000	37,400	37,400	7	

Areas of rounds taken from Carnegie's "Pocket Companion."

May 3, 1887.

EDGE MOOR IRON CO.,
By F. G. S.

ADDENDUM I. TABLE II. *l*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY MESSRS. E. AND T. FAIRBANKS & CO., OF NEW YORK.

Testing machine built by Fairbanks & Co. When procured, ... Maximum capacity, 200,000 lbs. How driven, hydraulic. Position of specimen (horizontal or vertical), vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			Dimensions before Test.			Length between Gauge Marks.						Stress in Lbs.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	Remarks.
			″	Area.	″	″										
1	W. Iron....	Square B.........	22	.986 .992 = .988	12	10	10.5	11.74	Not taken.	.801 .852 = .682	10.2	28,000	51,200	49,000	¼ hour each.	Elastic limit taken by W. H. Paine's gauge, reading to 1 ten-thousandth of an inch.
2	W. Iron....	Square B.........	22	.992 .988 = .980	12	10	10.1	13.02		.692 .675 = .467	9.4	30,000	52,570	46,000		Readings taken at every 2,000 lbs. until limit was reached, when gauge was removed, and specimen tested to destruction.
3	W. Iron	Square B.........	22	.994 .989 = .983	12	10	10.1	11.60		.791 .821 = .649	8.5	27,500	50,610	48,500		
4	W. Iron...	Flat Bar	22	2.523 .500 = 1.261	12	10	10.1	13.18		1.838 .327 = .601	11.8	44,000	70,420	62,860		
5	W. Steel...	Round B.........	18	.749 = .441	10	8	8.1	8.92		.541 = .230	6.5	24,000	35,750	28,000		
6	W. Steel...	Round B.........	18	.749 = .441	10	8	8.1	9.06		.536 = .226	6.8	23,000	34,850	28,220		
7	W. Steel...	Round B........	18	.749 = .437	10	8	8.1	8 80		.487 = .186	5.6	23,000	34,600	25,870		

Tested *July*, 1889.

N. O. OLSON,
84 Thomas St., New York City.

ADDENDUM I. TABLE II. *m*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY ALBERT W. STAHL, M.E., OF PURDUE UNIVERSITY, LA FAYETTE, IND.

Testing machine built by T. Olsen & Co., Philadelphia. When procured, October, 1885. Maximum capacity, 50,000 lbs. How driven, screw gearing, hand power. Position of specimen (horizontal or vertical), vertical.

															Min.	
168	Iron	Round	17	.4406	9.25	8.0	8.0157	8.737	.4394	.2561	5.0	56,741	77,392	59,328	90	
169	Steel	Round	17	.4418	9.40	8.0	8.0149	8.30	.4394	.1817	3.75	55,455	72,985	51,381	75	Fracture not between gauge marks.
170	Iron	Round	17	.4406	9.25	8.0	8.0166	8.85	.4394	.2173	6.37	57,875	75,805	59,239	70	
171	Iron	Round	17	.4394	9.25	8.0	8.0153	8.63	.4371	.2290	4.33	55,758	76,240	59,854	100	Fracture not between gauge marks.
172	Steel	Round	17	.4418	9.0	8.0	8.0182	8.52	.4394	.1932	4.68	67,904	80,806	56,360	100	
173	Steel	Round, sq. ends	22	.4453	11.0	8.0	8.0093	10.27	.4371	.2265	11.20	33,685	54,480	47,159	90	
174	Steel	Round, sq. ends	22	.4657	11.0	8.0	8.0093	8.83	.4608	.3959	6.25	28,989	49,130	49,130	60	
175	Steel	Round, sq. ends	22	.4815	11.0	8.0	8.0096	9.03	.4754	.3632	9.45	28,037	49,429	49,429	45	
176	Steel	Flat	22	.6165	14.5	8.0	8.0091	10.33	.6088	.2873	11.23	35,685		51,419	60	
177	Steel	Flat	22	.6373	14.5	8.0	8.0109	10.15	.6314	.3454	11.62	36,874	59,319	47,073	60	
													60,725			

January 26, 1887.

ALBERT W. STAHL.
La Fayette, Indiana.

BY PALMER C. RICKETTS, OF RENSSELAER POLYTECHNIC INSTITUTE, TROY, N. Y.

Testing machine built by Tinius Olsen. When procured, 1881. Maximum capacity, 50,000 lbs. How driven, hand. Position of specimen (horizontal or vertical), vertical.

Filed.				□″						□″					Min.	
2	Iron	Round	17	0.438	8.9	8	8.025	8.75	Not appreciably different from original section.	0.16	3.75	28,500	34,550	28,000	55	Fracture, fibrous non-crystalline.
3	Steel	Round	17	0.441	8.9	8	8.018	8.72		0.229	3.75	31,000	35,400	28,000	48	Fracture, dull $\frac{5}{8}$ cup.
4	Steel	Round	17	0.441	8.8	8	8.022	8.77		.281	7.50	29,500	34,300	24,700	31	Fracture, dull $\frac{6}{8}$ cup.
										0.1						

The shortness of the specimens sent prevented the use of a micrometer attachment for measuring strains, which may be read to the ten-thousandth of an inch and estimated to the hundred-thousandth. The wedges of the Olsen machine used require a grip of four inches at each end of the piece, which would have left but nine inches between grips, a distance too small to allow the micrometer attachment to measure the strain in eight inches. A grip of three inches only was tried on one of the iron specimens, but, as was feared, the balls slid in the sockets and the piece slipped in the wedges. The elastic limits of the remaining pieces were then determined by the use of compasses, the results being checked by the action of the scale beam.

As may be inferred from the above explanation, the Olsen machine used has wedge grips in ball-and-socket joints.

Brown & Sharp's vernier calipers, reading to the thousandth of an inch, were used for measuring diameters.

November 10, 1886.

PALMER C. RICKETTS.

ADDENDUM I. TABLE II. *n*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY PALMER C RICKETTS, OF TROY, N. Y.

Testing machine built by Tinius Olsen. When procured, 1881. Maximum capacity, 50,000 lbs. How driven, hand. Position of specimen (horizontal or vertical), vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			Dimensions before Test.			Length between Gauge Marks.						Stress in Lbs.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	Remarks.
				Mean Diam.						Diameter.					Min.	
5	Iron ..	Round...	22	0.7495	12.75	8	8.0149	8.54	*	0.562	4.4	28,500	36,300	28,000	38	Homog. fracture, No. crys. specks.
6	Iron ..	Round...	22	0.7479	12.75	8	8.0163	8.55	......	0.548	4.3	28,000	35,700	28,000	30	Homog. fracture, No. crys. specks.
7	Iron ..	Round...	22	0.7478	13.00	8	8.0164	8.56	......	0.525	4.3	26,500	35,900	28,500	27	Homog. fracture, No. crys. specks.
8	Steel..	Round...	22	0.7485	13.50	8	8.0155	8.87	......	0.503	4.3	26,500	36,600	26,500	30	Homog. frac. ¾ cup, medium depth.
9	Steel..	Round...	22	0.7483	13.50	8	8.0141	8.84	......	0.497	7.4	23,500	33,500	26,000	25	Homog. fracture, full cup, medium depth.
10	Steel..	Round...	22	0.7479	13.00	8	8.0142	8.87	......	0.488	5.7	24,000	32,300	24,800	22	Homog. frac. ¼ cup, medium depth.

* Considering the uncertainty of the point of fracture and the variation in the diameters of the bar at the same and at different sections, the reduction was not great enough to give decided results.

Micrometer arrangement was used to determine elastic limit readings to 0.0001 inch, estimations to 0.00001 inch. Ball-and-socket wedge grips used. Brown & Sharp's vernier micrometer used for sections: readings to 0.001 inch, estimations to 0.0001 inch.

January 15, 1887.

PALMER C. RICKETTS,
Troy, N. Y.

ADDENDUM I. TABLE II. *o.*

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY E. C. FELTON, ASSISTANT SUPERINTENDENT PENNSYLVANIA STEEL CO., OF STEELTON, PA.

Testing machine built by Tinius Olsen & Co. When procured, 1885. Maximum capacity, 100,000 lbs. How driven, by belt and Frisbie Friction Clutch. Position of specimen (horizontal or vertical), vertical.

		"			About	"									About	
1	Stee.	2½ × ½ Flat	....	2.517 × .510 1.2836 sq. in.	10 in.	8	Not observed.	10.32	Not observed.	2.000 × .382 .7640 sq. in.	Not observed.	48,100	74,600	Not observed.	5 min.	Bar very imperfect, having deep flaws running across bar about 4″ apart. Fracture occurred at one of these—silky.
2	Steel..	2½ × ½ Flat	....	2.522 × .524 1.3215 sq. in.	10 in.	8		10.80		1.850 × .342 .6327 sq. in.		49,800	75,550		5 min.	Fine silky cupped fracture.
1	Iron ..	¾ Round..	....	.750 diam. .4418 sq. in.	10 in.	8		8.20		.630 diam. .3117 sq. in.		33,100	35,000		5 min.	Fracture fibrous and ragged, showing piling of iron plainly. Broke in upper clamps.
2	Iron ..	¾ Round..	....	.749 diam. .4406 sq. in.	10 in.	8		8.70		.560 diam. .2463 sq. in.		32,000	35,700		5 min.	Broke outside of 8″ marks. Fracture fibrous, showing piling of iron.
1	Steel..	¾ Round..	....	.744 diam. .4347 sq. in.	10 in.	8		9.22		.500 diam. .1963 sq. in.		27,000	32,800		5 min.	Half-cup silky fracture.
2	Steel..	¾ Round..	....	.750 diam. .4418 sq. in.	10 in.	8		9.30		.485 diam. .1847 sq. in.		27,300	32,350		5 min.	Half-cup silky fracture.
1	Steel..	1 Square .	....	.993 × .996 .9890 sq. in.	10 in.	8		9.00		.808 × .850 .7068 sq. in.		34,000	48,550		5 min.	Fracture slightly cupped. Fine silky for about ⅛″ from surface, rest coarse granular, but not crystalline.
2	Steel..	1 Square .	....	1.000 × .996 .9960 sq. in.	10 in.	8		10.70		.625 × .635 .3968 sq. in.		32,000	51,250		5 min.	Well-marked cup, with center similar to No. 1.
3	Steel..	1 Square .	....	.996 × .997 .9930 sq. in.	10 in.	8		10.70		.644 × .640 .4117 sq. in.		33,700	51,200		5 min.	Fracture similar to No. 2. Fracture of square bars, fine silky, coarse granular.

NOTE.—The above tests were made in our regular way, which does not include any observation of elongation and reduction of area at elastic limit. Elastic limit determined by dropping of beam. ¾″ rounds of both iron and steel had been very badly "cold rolled."

November 8, 1886.

E. C. FELTON,
Assistant Superintendent Pennsylvania Steel Co.

ADDENDUM I. TABLE II. *p*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY TINIUS OLSEN & CO., OF PHILADELPHIA.

Testing machine built by Tinius Olsen & Co. When procured, new. Maximum capacity, 50,000 and 100,000 lbs. How driven, by hand for the first six specimens; by power for the last six specimens. Position of specimen (horizontal or vertical), vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			DIMENSIONS BEFORE TEST.			LENGTH BETWEEN GAUGE MARKS.						STRESS IN LBS.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	REMARKS.
			"		"	"	"	"			"				"	
1	Steel...	Round...	17	0.441 sq. in.	9	8	......	9.0	....	0.212 sq.in.	7	26,635	32,950	......	..	
2	Steel...	Round...	17	0.444 sq. in.	9	8	......	8.73	....	0.215 sq.in.	4	28,560	35,040	......	..	Broke outside of gauge marks for elongation.
3	Steel...	Round...	17	0.439 sq. in.	9	8	......	8.50	....	0.192 sq.in.	4.2	27,370	34,530	......	..	Broke outside of gauge marks for elongation.
4	Iron ...	Round...	17	0.441 sq. in.	9	8	......	9.0	....	0.239 sq.in.	6¼	22,300	34,780	......	..	
5	Iron ...	Round...	17	0.441 sq. in.	9	8	......	9.0	....	0.235 sq.in.	7	22,770	35,600	......	..	
6	Iron ...	Round...	17	0.442 sq in.	9	8	......	8.97	....	0.243 sq.in.	6¼	21,880	34,580	......	..	
1	Steel...	Square...	22	0.98 sq. in.	9	8	8.18	9.42	0.9506 sq.in.	0.6557 sq.in.	10¼	33,950	50,590	45,300	25	
2	Steel...	Square...	22	1.005 sq. in.	9	8	8.18	10.41	0.9552 sq.in.	0.4488 sq.in.	10	33,900	52,360	42,450	30	
3	Steel...	Square...	22	0.98 sq. in.	9	8	8.18	10.70	0.9506 sq.in.	0.4726 sq.in.	10	33,380	53,060	43,900	30	
1	Steel...	Flat......	22	2.52″ × 0.52″ 1.3104 sq. in.	9	8	8.17	10.35	1.275 sq. in.	0.7372 sq.in.	11	47,100	75,800	70,000	30	
2	Steel...	Flat......	22	2.52″ × 0.52″ 1.3104 sq. in.	9	8	8.18	10.35	1.275 sq. in.	0.6545 sq.in.	9¼	47,800	75,220	66,200	30	
3	Steel...	Flat......	22	2.52″ × 0.52″ 1.3104 sq. in.	9	8	8.18	10.55	1.275 sq. in.	0.6545 sq.in.	9	48,200	75,240	66,570	30	

May 4, 1887.

TINIUS OLSEN & CO.,
per T. O.

ADDENDUM I. TABLE II. *q*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY PITTSBURGH TESTING LABORATORY, OF PITTSBURGH, PA.

Testing machine built by Messrs. Tinius Olsen & Co. When procured, 1881–86. Maximum capacity, see remarks. How driven, see remarks. Position of specimen (horizontal or vertical), vertical.

			"		"	"		"			"				Min.	
2339	Steel...	¾" Round	17	.748 diam.	10	8	8.02	8.86	.748 diam.	.521 diam.	8.	25,000	34,000	26,300	35	Tests made on a 50,000 lbs. machine. Driven by a 1¼ H.P. electric motor. 1881.
2340	Steel...	¾" Round	17	.746 diam.	10	8	8.03	8.84	.746 diam.	.500 diam.	5.25	27,000	36,460	24,700	40	
2341	Steel...	¾" Round	17	.748 diam.	10	8	8.02	8.88	.748 diam.	.550 diam.	5.0	25,000	34,590	27,000	30	
2342	Iron ...	¾" Round	17	.745 diam.	10	8	8.04	8.94	.745 diam.	.547 diam.	9.0	25,000	35,250	28,000	30	
2343	Iron ...	¾" Round	17	.745 diam.	10	8	8.03	9.00	.745 diam.	.548 diam.	8.5	25,000	34,190	26,400	30	
2344	Steel...	1" Square	22	1.009 × .975	7¾	8	8.08	10.62	1.009 × .975	.635 × .653	12.0	33,080	51,860	40,000	16	Tests made on a 200,000 lbs. machine. Driven by a 50 H.P. steam-engine. 1886.
2345	Steel...	1" Square	22	1.010 × .974	8⅝	8	8.04	10.64	1.010 × .974	.645 × .667	11.25	33,600	51,540	40,000	20	
2346	Steel...	1" Square	22	.990 × .990	8⅝	8	8.02	10.62	.990 × .990	.633 × .640	11.75	31,880	51,460	39,550	15	
2347	Steel...	2½" × ½"	22	2.520 × .499	9¼	8	8.04	10.52	2.520 × .499	1.885 × .353	10.	46,530	69,500	60,540	15	
2348	Steel...	2½" × ½"	22	2.520 × .498	9	8	8.02	10.53	2.520 × .498	1.940 × .372	11.0	49,240	75,490	66,400	25	

January, 1887.

PITTSBURGH TESTING LABORATORY, HUNT & CLAPP.

ADDENDUM I. TABLE II. *r*.

REPORT OF TENSION TESTS FOR THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

BY A. AND P. ROBERTS & CO., OF PENCOYD IRON WORKS, PENCOYD.

Testing machine built by Tinius Olsen & Co. When procured, August, 1887. Maximum capacity, 200,000 lbs. How driven, steam power. Position of specimen (horizontal or vertical), vertical.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
			Dimensions before Test.			Length between Gauge Marks.						Stress in Lbs.				
No. or mark of specimen.	Kind of material.	Form of specimen.	Total length.	Section.	Length between grips or jaws when starting test.	When starting test.	At elastic limit while under strain.	After rupture.	Minimum section at elastic limit.	Minimum section after fracture.	Distance of point of fracture from nearest end of specimen.	At elastic limit.	Maximum observed.	At time of rupture.	Duration of test.	Remarks.
			″		″	″		″			″				Min.	
1359	$\frac{3}{4}$″ Steel.	Round	17	.747 diam.	6.5	8	......	9.	......	.524 diam.	4.2	27,110	31,730	23,650	15	Fibrous.
1260	$\frac{3}{4}$″ Steel.	Round	17	.747 diam.	4.75	8	......	8.74	......	.500 diam.	7.2	26,200	37,150	26,180	12	Fibrous.
1361	$\frac{3}{4}$″ Steel.	Round	17	.744 diam.	5.	8	......	8.67	......	.524 diam.	5.75	25,710	34,900	26,900	16	Fibrous.
1362	$\frac{3}{4}$″ Iron .	Round	17	.747 diam.	5.	8	......	9.00	......	.550 diam.	6.2	16,630	35,300	25,500	13	Fibrous.
1363	$\frac{3}{4}$″ Iron .	Round	17	.744 diam.	5.	8	......	9.01	......	.555 diam.	7.7	19,260	34,300	27,200	15	Fibrous.
1364	$\frac{3}{4}$″ Iron .	Round	17	.747 diam.	4.5	8	......	8.96	......	.554 diam.	7.0	18,150	35,860	28,760	14	Fibrous.
1365	Steel	Square	22	.990 × .988	11.25	8	......	10.64	......	.596 × .607	10.38	31,100	50,100	39,150	18	Fibrous.
1366	Steel	Square	22	1.010 × .974	11.00	8	......	10.63	......	.624 × .663	12.5	31,520	50,530	40,000	20	Fibrous.
1367	Steel	Square	22	.990 × .992	11.50	8	......	10.73	......	.634 × .636	11.5	30,800	49,810	39,800	19	Fibrous.
1368	Steel ...	$2\frac{1}{2}$″ × $\frac{1}{2}$″ Flat.	22	2.515 × .527	11.00	8	......	10.75	......	1.819 × .324	11.7	47,740	73,720	62,300	25	Fibrous.
1369	Steel	$2\frac{1}{2}$″ × $\frac{1}{2}$″ Flat.	22	2.515 × .502	10.00	8	......	10.51	......	1.870 × .338	12.7	46,400	66,500	58,000	25	Rough fracture.
1370	Steel	$2\frac{1}{2}$″ × $\frac{1}{2}$″ Flat.	22	2.515 × .504	9.50	8	......	10.59	......	1.813 × .320	10.6	45,950	67,470	57,570	25	Rough fracture.

November 19, 1887.

NAT. EWING,
Pencoyd.

RÉSUMÉ OF RESULTS—ADDENDUM I. TABLE II. s.

Name.	Round Steel.			Square Steel.			Round Iron.			Flat Iron.			Name of Testing Machines used.
	Elastic limit.	Tenacity.	Elongation.	Elastic limit.	Tenacity.	Elongation.	Elastic limit.	Tenacity.	Elongation.	Elastic limit.	Tenacity.	Elongation.	
Cambria Iron Co.	57,000	74,550	12.62	31,000	50,100	18.10	59,000	77,930	8.62	37,240	54,400	32.37	6 tests of rounds by Cambria on *Gill*.
	64,000	80,580	6.25	31,500	52,330	31.5	59,000	77,450	11.6	37,900	58,900	31.81	
	70,000	82,480	6.37	34,000	52,440	31.9	59,000	78,680	6.75	36,666	53,500	33.25	
Ordnance Department, U. S. A., Watertown Arsenal	....	81,160	25.87	33,870	52,020	21.0	....	84,440	3.87	36,950	57,560	33.25	
	....	86,220	7.37	31,690	54,500	32.75	....	87,220	10.5	36,900	53,920	32.87	
	....	83,920	4.00	33,000	52,480	33.5	....	81,220	3.62	36,570	57,140	29.87	
Washburn & Moen Manufacturing Co.	74,710	86,040	10.50	....	....	....	65,810	79,430	7.00	....	....	....	*Emery*.
	63,400	74,710	12.87	....	....	....	50,480	77,600	12.37	....	....	....	
	58,950	72,550	13.37	....	....	....	63,630	78,220	13.62	....	....	....	
Yale & Towne Manufacturing Co.	27,440	85,[illegible]60	9.98	35,011	52,320	31.6	23,940	81,040	8.25	38,030	60,170	30.00	
	36,440	82,860	11.25	33,550	52,660	26.6	24,720	78,540	....	38,960	60,460	30.00	
	36,210	54,660	....	33,200	51,520	33.1	24,900	79,910	....	38,950	60,370	30.00	
Pottsville Iron and Steel Co.	66,090	88,290	13.25	....	....	....	72,300	81,990	24.12	....	....	....	
	62,470	73,800	12.12	....	....	....	67,260	83,820	12.00	....	....	....	
	71,820	83,290	11.50	....	....	....	63,880	91,100	12.37	....	....	....	
Chester Rolling Mills	64,520	86,030	7.18	34,610	52,850	31.85	40,500	78,940	12.00	36,300	59,110	28.56	*Riehlé*.
	61,440	69,690	13.10	33,850	52,100	32.50	50,000	81,380	12.71	38,430	60,980	25.72	
	63,720	73,960	5.00	32,100	48,230	15.7	64,860	30,350	12.71	38,180	60,760	26.46	
Riehlé Bros.	61,440	69,970	....	32,980	52,980	32.50	63,070	79,530	8.75	37,710	60,260	31.75	
	68,500	76,970	14.5	32,460	52,130	35.90	62,710	76,610	10.00	37,900	58,750	....	
	64,760	75,670	16.25	32,470	52,650	33.50	62,500	79,970	12.50	38,720	57,730	32.12	

RÉSUMÉ OF RESULTS—ADDENDUM I. TABLE II. *t*.

Name.	Round Steel.			Square Steel.			Round Iron.			Flat Iron.			Name of Testing Machines Used.
	Elastic limit.	Tenacity.	Elongation.	Elastic limit.	Tenacity.	Elongation.	Elastic limit.	Tenacity.	Elongation.	Elastic limit.	Tenacity.	Elongation.	
Sibley College, Cornell University	63,501	77,232	11.25	33,030	54,107	24.25	57,088	76,894	10.50	38,045	57.768	22.37	*Fairbanks*
	64,987	76,685	10.75	33,030	54,159	23.00	54.729	75,481	10.37	38,669	57,768	21.75	
	....	....	....	33,030	54,562	23.25	....	....	....	....	....	....	
Edge Moor Iron Co.	46,800	78,690	10.37	34,600	53,460	30.27	47,030	78,920	12.50	38,130	59,310	26.25	
	51,620	86,020	8.12	33,820	51,940	30.00	43,440	81,440	12.37	38,400	55,990	28.50	
	52,760	85,800	10.62	34,140	51,920	32.50	44,970	78,960	12.12	36,740	55,670	28.12	
Fairbanks & Co.	54,420	81,080	11.50	28,340	65,250	17.4	....	....	....	34,900	55,850	31.80	
	52,160	79,030	13.25	30,540	53,650	30.20	....	....	....	....	....	....	
	52,620	79,160	10.00	27,970	51,490	16.0	....	....	....	....	....	....	
Purdue University	55,455	72,985	3.75	33,685	54,480	28.37	56,741	77.392	9.21	35,685	59,319	29.10	*Olsen.*
	67,904	80,806	6.50	28,989	49,130	10.37	57,875	75.805	10.62	36,874	60,725	26.90	
	....	....	....	28,037	49,429	12.90	55,758	76,240	7.90	....	....	....	
Rensselaer Polytechnic Institute	60,230	83,200	10.87	....	....	....	64.610	82,300	6.75	....	....	....	
	53,450	74.430	10.50	....	....	....	63,810	81,320	6.62	....	....	....	
	51,010	73,560	10.87	....	....	....	60,340	81,760	7.00	....	....	....	
Pennsylvania Steel Co.	62.110	75,450	15.25	34.380	49,080	12.50	74,900	79,230	2.50	37,500	58,150	29.00	
	61,800	73,200	16.25	32,120	51,450	33.75	72,610	81,020	8.75	37,400	56,750	35.00	
	....	....	....	53,940	51,570	33.75	....	....	....	....	....	....	
Tinius Olsen & Co.	60,410	74,720	12.50	34.640	51,640	17.75	50,560	78,880	12.50	35,950	57,870	29.37	
	64,310	78,920	9.12	33,750	52,180	30.12	51,640	80,720	12.50	36,480	57,410	29.37	
	62,350	78,660	6.25	34,050	54,150	33.75	49,510	78,240	12.12	36,780	57,430	31.87	
Pittsburgh Testing Laboratory	56,890	77,360	10.50	33,630	52,710	32.39	57,340	80,850	11.25	37,030	55,300	31.00	
	61,770	83,400	10.12	34,150	53,380	32.50	57,090	78,440	12.12	39,250	59,120	31.37	
	56,890	78,720	10.75	32,520	52,520	32.50	....	....	....	....	....	....	
Pencoyd Iron Works	61,870	72,420	12.50	31,510	50,750	33.00	37,940	80.550	12.50	36,030	55,650	34.37	
	59,780	84,780	9.25	32,050	51.380	32.87	44,290	78,900	12.62	36,760	5,269	31.37	
	59,140	80,260	8.37	31,370	50,720	34.12	41,410	81,820	12.00	36,280	5,326	32.37	

NOTE.

This report, presented as one of the papers of the Cincinnati Convention of the Society, in May, 1890, is printed here for circulation among those interested, and with a view to elicit discussion and suggestion. When such criticism has been received and considered, the report, with appended discussion, will be presented for acceptance as the final views of the Committee, to receive the usual action of the Society upon reports of this kind.

Secretary.

42

CCCLXXXI.

REPORT OF COMMITTEE ON A STANDARD METHOD OF CONDUCTING DUTY TRIALS OF PUMPING ENGINES.

To the American Society of Mechanical Engineers:

The committee of five, who were appointed at the Nashville meeting of the Society, held in May, 1888, to determine upon a standard method of conducting duty trials of steam pumping engines, have endeavored to carry out the work intrusted to them, and they beg leave to report upon the same as follows:

1. The need of a uniform system of determining the performance of pumping engines is widely recognized, and it has already been commented upon at such length in the Paper and Discussion which led to the appointment of the committee, that but little requires to be added here. The main objects to be secured by the proposed standard method seem to be, first, to establish, for the benefit of members of the Society, and of others who care to use it, a mode of determining the performance of pumping engines, which may guide them in making tests themselves, and which, in contracts between builder and purchaser, may be specified as the mode to be followed in determining whether or not the guaranteed duty of the engine is realized; and, second, to furnish a common basis on which to compare the economy of different engines.

2. The committee has taken it for granted that the scope of its work extends over the whole field of duty trials, and that it is not confined simply to devising a suitable method for carrying on the operations of making the test.

It requires only a brief examination of the subject to see that much of the present variety of results which are obtained, and which it is desirable to overcome, is due to the varied nature of the coal unit upon which the duty is now based. In the eastern section of our country the unit of "100 lbs. of coal" may refer either to Cumberland bituminous coal or to anthracite coal, and in the middle or western section to Pittsburg, Illi-

nois, or Ohio bituminous coal. In some localities the fuel may be petroleum, natural gas, coal screenings, tan-bark, or sawdust. There is a wide difference in the evaporative efficiency of these various fuels, even when of good quality, and differences of quality in the same grade of coal are the cause of still greater diversity. These variations in kind and quality of fuel, which are significant in no small degree, make the old standard unfit either for a commercial or a scientific basis for duty ratings. It is proposed, therefore, at the outset, that the existing unit, "100 lbs. of coal," be abolished, and that, in its place, a new basis, "1,000,000 heat units," be established. One million heat units is a quantity of heat corresponding to that produced in the combustion of 100 lbs. of coal which gives out 10,000 heat units per pound, or which produces an evaporation of $\frac{10,000}{965.7} = 10.355$ lbs. of water from and at 212 degrees per pound of fuel. This evaporative result is readily obtained from 100 lbs. of all grades of Cumberland bituminous coal, used in horizontal return tubular boilers, and, in many cases, from 100 lbs. of the best grades of anthracite coal. The proposed new unit is thus, in reality, though not in name, in close accord with the existing unit, and, furthermore, it retains its numerical simplicity.

3. Considering that the two processes by which steam is generated and used are wholly distinct and independent of each other, there is a natural line of separation between the work of the boiler and that of the engine, and it is impossible to determine their individual economy if treated as a whole. There is reason here for separating the performance of the engine, in the proposed standard, from that of the boilers, and this mode of independent treatment is the one which the committee recommends. An additional reason for this course is found in the somewhat extended practice, among those who purchase pumping engines, of obtaining the boilers from independent builders.

In order that a contract may be brought into accord with this provision, where the complete plant is furnished by one party, it would be framed with two clauses relating to the performance, one referring to the duty of the engine proper, and the other referring to the evaporative duty of the boilers.

4. Starting with a heat-unit basis of computing duty, it is proposed to make the computation from the quantity of heat supplied to the complete plant; using not only that supplied to the

engine cylinders, but that supplied to all the accessory parts of the engine, such as the steam-jackets, the donkey feed-pump, the independent air-pump, if this be driven with steam, and any other apparatus using steam which is necessary to the operation of the engine. It is recommended that the scope of the test be made so broad that, for the sake of completeness, the quantity of steam which passes through the cylinders of the engine be determined independently of that used for other purposes, and likewise, that the quantity of steam used by each accessory part of the engine be also determined. In contract tests, if a steam-pump be used for the boiler feed-pump, the quantity of heat supplied for operating this apparatus is to be included in the total quantity, not only in cases where both boiler and engine are supplied by one party, but also where the boiler is furnished by a separate contractor. In this connection it should be added that if the engine contractor does not furnish the boiler feed-pump, he should be permitted to specify, if he desires, the kind of feeding apparatus which shall be used during the test.

The heat-unit method requires that the actual total heat of the steam shall be known, and for this purpose allowance will necessarily be made for any moisture or superheat contained by the steam furnished to the engine.

5. In determining upon a suitable method of measuring the amount of work done, which involves a measure of the quantity of water discharged into the force main, the committee have endeavored to find one which may be employed universally, and which may, in a reasonable manner, serve the ends of the builder, purchaser, and all interested parties. Plunger displacement and weir measurement have heretofore been the common means of ascertaining the quantity of water discharged. The use of a Venturi tube, so called, inserted in the force main, has been advocated, as also the employment of nozzles, or other similar means of indirect determination. [See Appendix.]

The objection to the plunger-displacement method is that, on account of "slip," which, as the committee understand, covers all the losses due to leakage of plunger and valves, to the return of water through the valves during the interval of closing, and to imperfect filling of the pump, the quantity calculated is greater than the actual discharge, and the method does not afford, in contract tests, the protection to the purchaser which his interest demands. The objection to the indirect modes of measurement,

as by the use of weir, tube, or nozzles, is that the person making the test must educate himself in the manipulation of the apparatus chosen, so as to be himself assured of the reliability of the data; and, furthermore, whether thus assured or not, he must compute the desired quantities by the use of coefficients, the accuracy of which he cannot himself verify, and which may not exactly apply to the precise conditions relating to the individual case in hand. There is no method thus far used or advocated which is not open to some kind of objection, and the committee are obliged to recommend a course which, though subject to criticism, appears to reduce the objectionable features to a minimum.

It is proposed that the plunger-displacement system of measurement be employed, and that the purchaser's interest be protected by the determination of the amount of slip in the pump, so far as slip is produced by leakage of the plunger, which is probably its main factor, and leakage of valves, if this occurs through faulty design. It may be said that, if it were not for leakage, the pump itself, in well-constructed engines, would undoubtedly furnish the most reliable meter which could be had of the quantity of water discharged. A satisfactory determination of the approximate extent to which leakage occurs does not present serious difficulty.

In deciding upon plunger displacement, accompanied by a determination of the leakage, as the best mode of measurement for the purposes in view, the committee does not for a moment underrate the importance and desirability of measurement by weir, tube, or nozzle, whenever either of these can be employed to advantage. It is strongly recommended that these additional measurements be undertaken in all cases where it is practicable to do so, that the results of the test may be supplemented by the additional data thus obtained. [See Appendix.]

6. In determining the quantity of work done by the pump, the committee recommends that the work of overcoming the friction of the water in passing through the passages and valves in the pump should not be included in the desired total; but that the work expended in friction of both the force and suction mains be included in that on which the duty is computed. It is held that the efficiency of the engine should not be made dependent upon any condition which is foreign to itself, and that the builder of the engine should be held responsible only for the work done

from the time when the water enters the pump to the time when it leaves it. The purchaser, it should be observed, should guard his interest in the matter by having the mains furnished of such capacity as to reduce their friction to a minimum.

To carry out these provisions, the data to be determined, apart from that relating to the plunger displacement, are the indication of a pressure gauge attached to the force main, that of a vacuum gauge attached to the suction main, and the vertical distance between the centres of the two gauges.

It is recommended that no air be allowed to enter the pump cylinders during the progress of the test, thereby removing all possibility of imperfect filling. If it is necessary, in special cases, for air to be "snifted in," this should be regarded as a defect in the action of the pump, which should be noted by the expert in his report, and such allowance should be made for the imperfect filling, due to the presence of air, as may be determined upon by an examination of the indicator diagrams taken from the pump cylinders, or from other data which may be secured.

7. The necessary data having been obtained in accordance with these recommendations, the duty of an engine, and other quantities relating to its performance, may be computed by the use of the following formulæ:

1. $$\text{Duty} = \frac{\text{Foot pounds of work done}}{\text{Total number of heat units consumed}} \times 1{,}000{,}000$$

$$= \frac{A\,(P \pm p + s) \times L \times N}{H} \times 1{,}000{,}000 \text{ (foot pounds).}$$

2. $$\text{Percentage of leakage} = \frac{C \times 144}{A \times L \times N} \times 100 \text{ (per cent.).}$$

3. Capacity = number of gallons of water discharged in 24 hours

$$= \frac{A \times L \times N \times 7.4805 \times 24}{D \times 144}$$

$$= \frac{A \times L \times N \times 1.24675}{D} \text{ (gallons).}$$

4. Percentage of total frictions

$$= \left[\frac{I.H.P. - \dfrac{A\,(P \pm p + s) \times L \times N}{D \times 60 \times 33{,}000}}{I.H.P.}\right] \times 100$$

$$= \left[1 - \frac{A\,(P \pm p + s) \times L \times N}{A_s \times M.E.P. \times L_s \times N_s}\right] \times 100 \text{ (per cent.);}$$

or, in the usual case, where the length of the stroke and number of strokes of the plunger are the same as that of the steam-piston, this last formula becomes:

$$\text{Percentage of total frictions} = \left[1 - \frac{A\,(P \pm p + s)}{A_s \times M.E.P.}\right] \times 100 \text{ (per cent.)}.$$

In these formulæ, the letters refer to the following quantities:

A = Area, in square inches, of pump plunger or piston, corrected for area of piston-rod. (When one rod is used at one end only, the correction is one-half the area of the rod. If there is more than one rod, the correction is multiplied accordingly.)

P = Pressure, in pounds per square inch, indicated by the gauge on the force main.

p = Pressure, in pounds per square inch, corresponding to indication of the vacuum gauge on suction main (or pressure gauge, if the suction pipe is under a head). The indication of the vacuum gauge, in inches of mercury, may be converted into pounds by dividing it by 2.035.

s = Pressure, in pounds per square inch, corresponding to distance between the centres of the two gauges. The computation for this pressure is made by multiplying the distance, expressed in feet, by the weight of one cubic foot of water at the temperature of the pump well, and dividing the product by 144; or by multiplying the distance in feet by the appropriate quantity, found in the following table. The quantities in this table are computed from the weights of one cubic foot of water at the various temperatures, as given by D. K. Clark in his *Rules and Tables*, which also correspond to Charles T. Porter's figures, in his work on the *Richards' Steam-Engine Indicator*.

Temperature of Water in Pump Well.	Weight of 1 cu. ft. of Water divided by 144.	Temperature of Water in Pump Well.	Weight of 1 cu. ft. of Water divided by 144.
Deg. Fahr.		Deg. Fahr.	
32	.4335	75	.4325
35	.4335	80	.4322
40	.4335	85	.4319
45	.4334	90	.4315
50	.4333	95	.4311
55	.4332	100	.4307
60	.4331	105	.4303
65	.4329	110	.4298
70	.4327		

L = Average length of stroke of pump plunger, in feet.

N = Total number of single strokes of pump plunger made during the trial.

A_s = Area of steam-cylinder, in square inches, corrected for area of piston-rod. The quantity, $A_s \times M.E.P.$ in an engine having

more than one cylinder, is the sum of the various quantities relating to the respective cylinders.

L_s = Average length of stroke of steam-piston, in feet.

N_s = Total number of single strokes of steam-piston during trial.

$M.E.P.$ = Average mean effective pressure, in pounds per square inch, measured from the indicator diagrams taken from the steam-cylinder.

$I.H.P.$ = Indicated horse-power developed by the steam-cylinder.

C = Total number of cubic feet of water which leaked by the pump plunger during the trial, estimated from the results of the leakage test.

D = Duration of trial, in hours.

H = Total number of heat units [$B.T.U.$] consumed by engine = weight of water supplied to boiler by main feed-pump × total heat of steam of boiler pressure reckoned from temperature of main feed-water + weight of water supplied by jacket-pump × total heat of steam of boiler pressure reckoned from temperature of jacket-water + weight of any other water supplied × total heat of steam reckoned from its temperature of supply. The total heat of the steam is corrected for the moisture or superheat which the steam may contain. For moisture, the correction is subtracted, and is found by multiplying the latent heat of the steam by the percentage of moisture, and dividing the product by 100. For superheat, the correction is added, and is found by multiplying the number of degrees of superheating (*i.e.*, the excess of the temperature of the steam above the normal temperature of saturated steam) by 0.48. No allowance is made for heat added to the feed-water, which is derived from any source, except the engine or some accessory of the engine. Heat added to the water by the use of a flue heater at the boiler is not to be deducted. Should heat be abstracted from the flue by means of a steam reheater connected with the intermediate receiver of the engine, this heat must be included in the total quantity supplied by the boiler.

The total and latent heats may be found by reference to the Tables of the Properties of Saturated Steam, given in Table No. 1 of the Appendix. The quantities of heat contained in one pound of water at various temperatures are appended in Table No. 2. These Tables are copied from Charles T. Porter's treatise on the *Richards' Steam-Engine Indicator*. The pressures here given are reckoned from zero. To convert the gauge pressure to that referred to as the zero basis, the barometric pressure is to be added to the corrected indications of the gauge. When the barometer indicates 29.92 inches, the pressure to be added is 14.7 lbs. per square inch. For other indications of the barometer, the corresponding pressure may be found by using the multiplier 0.491.

The following examples are given to illustrate the method of computation. The figures are not obtained from tests actually

made, but they correspond in round numbers with those which were so obtained :

First Example.—Compound tandem direct-acting duplex engine. Both high-pressure and low-pressure cylinders jacketed with live steam. Jet condenser used, with air-pump driven by main engine. Boiler feed-pump also driven by main engine. Jacket water returned to boiler by gravity. Main supply of feed-water drawn from hot well.

DIMENSIONS.

Diameter of each high-pressure cylinder (two).......	15	ins.
Diameter of each low-pressure cylinder (two)........	30	"
Diameter of piston-rod, each cylinder (one at each end high-pressure, two at one end low-pressure) ..	3.5	"
Diameter of pump plungers (two)....................	15	"
Diameter of piston-rod, each plunger (one at one end)	3.5	"
Nominal stroke..................................	18	"

GENERAL DATA.

1.	Duration of test (D)...........	12	hrs.
2.	Boiler pressure by gauge (barometric pressure 14.7 lbs.)..............................	120	lbs.
3.	Temperature of water in pump well..........	80°	
4.	Temperature of main supply of feed-water....	100°	
5.	Temperature of jacket water........	280°	
6.	Percentage of moisture in steam.............	0%	
7.	Weight of water supplied to boiler by main feed-pump........	22,400	lbs.
8.	Weight of water supplied to boiler by jackets.	2,560	"

DATA RELATING TO WORK OF PUMP.

9.	Area of plunger minus ½ area of rod (A).	171.9	sq. ins.
10.	Average length of stroke (L and L_s)...	1.572	ft.
11.	Total number of single strokes during trial (N and N_s)....................	76,000	
12.	Pressure by gauge on force main (P)...	100	lbs.
13.	Vacuum by gauge on suction main....	9.3	ins.
14.	Pressure corresponding to vacuum given in preceding line (p)	4.57	lbs.
15.	Vertical distance between gauges......	8	ft.
16.	Pressure corresponding to distance given in preceding line (s)...........	3.46	lbs.
17.	Volume of water which leaked through the plungers computed from results of leakage test (C)..................	5,900	cu. ft.

DATA RELATING TO WORK OF STEAM-CYLINDERS.

18.	Area of high-pressure piston minus area of one rod (A_{s_1})......	167.09	sq. ins.

19. Mean effective pressure high-pressure cylinder ($M.E.P._1$)	61.31	lbs.
20. Area of low-pressure piston minus ½ area of two rods (A_{s_2})...............	697.24	sq. ins.
21. Mean effective pressure low-pressure cylinder ($M.E.P._2$)................	13.72	lbs.
22. Number of double strokes each side per minute	26.39	
23. Indicated horse-power developed by steam-cylinders	99.61	I.H.P.
24. Feed-water consumed per indicated horse-power per hour	20.88	lbs.
25. Heat units consumed per indicated horse-power per hour............... = 383 B.T.U. per minute.	23,008	B.T.U.

TOTAL HEAT OF STEAM RECKONED FROM THE VARIOUS TEMPERATURES OF FEED-WATER AND COMPUTATIONS BASED THEREON.

26. Total heat of 1 lb. of dry steam at 120 lbs. gauge pressure reckoned from 0° Fahr..................	1,220.6	B.T.U.
27. Ditto, reckoned from temperature of main feed-water (100°)	1,120.5	"
28. Ditto, reckoned from temperature of jacket-water (280°)................	938.5	"
29. Heat consumed by engine (H) (22,400 × 1120.5) + (2560 × 938.5)..........	27,501,760	"

RESULTS.

Substituting these quantities in the formulæ, we have :

$$1.\ \text{Duty} = \frac{\overset{A}{171.9} \times (\overset{P}{100} + \overset{p}{4.57} + \overset{s}{3.46}) \times \overset{L}{1.572} \times \overset{N}{76,000}}{\overset{H}{27,501,760}} \times 1,000,000$$

= 80,671,622 foot pounds.

$$2.\ \text{Percentage of leakage} = \frac{\overset{C}{5,900} \times 144}{\overset{A}{171.9} \times \overset{L}{1.572} \times \overset{N}{76,000}} \times 100 = 4.1\%.$$

$$3.\ \text{Capacity} = \frac{\overset{A}{171.9} \times \overset{L}{1.572} \times \overset{N}{76,000} \times 1.24675}{\overset{D}{12}}$$

= 2,133,735 gallons.

4. Percentage of total frictions

$$= \left[1 - \frac{\overset{A}{171.9} \times (\overset{P}{100} + \overset{p}{4.57} + \overset{s}{3.46})}{(\overset{A_{s_1}}{167.09} \times \overset{M.E.P._1}{61.31}) + (\overset{A_{s_2}}{697.24} \times \overset{M.E.P._2}{13.72})}\right] \times 100$$

= 9.4%.

Second Example.—Compound fly-wheel engine. High-pressure cylinder jacketed with live steam from the boiler. Low-pressure cylinder jacketed with steam from the intermediate receiver, the condensed water from which is returned to the boiler by means of a pump operated by the engine. Main steam-pipe fitted with a separator. The intermediate receiver provided with a reheater supplied with boiler steam. Water drained from high-pressure jacket, separator, and reheater, collected in a closed tank under boiler pressure, and from this point fed to the boiler direct by an independent steam-pump. Jet condenser used operated by an independent air-pump. Main supply of feed-water drawn from hot well and fed to the boiler by donkey steam-pump, which discharges through a feed-water heater. All the steam-pumps, together with the independent air-pump, exhaust through the heater to the atmosphere.

DIMENSIONS.

Diameter of high-pressure steam-cylinder (one)........	20	ins.
Diameter of low-pressure steam-cylinder (one).........	40	"
Diameter of plunger (one).........................	20	"
Diameter of each piston-rod	4	"
Stroke of steam-pistons and pump plunger............	3	ft.

GENERAL DATA.

1.	Duration of trial (D)......................	10	hrs.
2.	Boiler pressure indicated by gauge (barometric pressure, 14.7 lbs.)..............	120	lbs.
3.	Temperature of water in pump well	60°	
4.	Temperature of water supplied to boiler by main feed-pump, leaving heater	215°	
5.	Temperature of water supplied by low-pressure jacket pump....	225°	
6.	Temperature of water supplied by high-pressure jacket, separator, and reheater pump, that derived from separator being 340°, and that from jackets 290°.........	300°	
7.	Weight of water supplied to boiler by main feed-pump	18,863	lbs.
8.	Weight of water supplied by low-pressure jacket pump...........................	615	"
9.	Weight of water supplied by pump for high-pressure jacket, separator, and reheater tank, of which 210 lbs. is derived from separator........................	1,025	"
10.	Total weight of feed-water supplied from all sources	20,503	"
11.	Percentage of moisture in steam after leaving separator.........................	1.5%	

DATA RELATING TO WORK OF PUMP.

12.	Area of plunger minus ½ area of piston-rod (A)	307.88	sq. ins.
13.	Average length of stroke (L and L_2)...	3	ft.

14. Total number of single strokes during trial (N and N_s)	24,000	
15. Pressure by gauge on force main (P)	95	lbs.
16. Vacuum by gauge on suction main	7.5	ins.
17. Pressure corresponding to vacuum given in preceding line (p)	3.69	lbs.
18. Vertical distance between centres of two gauges	10	ft.
19. Pressure equivalent to distance between two gauges (s)	4.33	lbs.
20. Total leakage of pump during trial, determined from results of leakage test (C)	3,078	cu. ft.
21. Number of double strokes of pump per minute	20	
22. Mean effective pressure measured from pump diagrams	105	lbs.
23. Indicated horse-power exerted in pump cylinders	117.55	I.H.P.

DATA RELATING TO WORK OF STEAM-CYLINDERS.

24. Area of high-pressure piston minus ½ area of rod (A_{s1})	307.88	sq. ins.
25. Area of low-pressure piston minus ½ area of rod (A_{s2})	1,250.36	"
26. Average length of stroke, each	3	ft.
27. Mean effective pressure measured from high-pressure diagrams ($M.E.P._1$)	59.25	lbs.
28. Mean effective pressure measured from low-pressure diagrams ($M.E.P._2$)	13.60	"
29. Number of double strokes per minute (line 21)	20	
30. Indicated horse-power developed by high-pressure cylinder	66.33	I.H.P.
31. Indicated horse-power developed by low-pressure cylinder	61.82	"
32. Indicated horse-power developed by both cylinders	128.15	"
33. Feed-water consumed by plant per indicated horse-power per hour, corrected for separator water and for moisture in steam	15.60	lbs.
34. Number of heat units consumed per indicated horse-power per hour	15,652.1	B.T.U.
35. Number of heat units consumed per indicated horse-power per minute	260.9	"

TOTAL HEAT OF STEAM RECKONED FROM THE VARIOUS TEMPERATURES OF FEED-WATER AND COMPUTATIONS BASED THEREON.

36. Total heat of 1 lb. of steam at 120 lbs. gauge pressure, containing 1.5% of

moisture, reckoned from 0° Fahr. = 1,220.6 − (1.5% of 866.7) .... 1,207.6 B.T.U.

37. Ditto, reckoned from 215°, temperature of main feed-water = 1,207.6 − 215.9 .... 991.7 "

38. Ditto, reckoned from 225°, temperature of low-pressure jacket water = 1,207.6 − 226.1 .... 981.5 "

39. Ditto, reckoned from 290°, temperature of high-pressure jacket and reheater water = 1,207.6 − 292.3 ... 915.3 "

40. Heat of separator water reckoned from 340° = 353.9 − 343.8................ 10.1 "

41. Heat consumed by engine (H) = (18.863 × 991.7) + (615 × 981.5) + (815 × 915.3) + (210 × 10.1)................ 20,058,150 "

RESULTS.

Substituting these quantities in the formulæ, we have:

$$1.\ \text{Duty} = \frac{\overset{A}{307.88} \times (\overset{P}{95} + \overset{p}{3.69} + \overset{s}{4.33}) \times \overset{L}{3} \times \overset{N}{24,000}}{\underset{20,058,150}{H}} \times 1,000,000$$

$$= 113,853,044 \text{ foot pounds.}$$

$$2.\ \text{Percentage of leakage} = \frac{\overset{C}{3,078 \times 144}}{\overset{A}{307.88} \times \overset{L}{3} \times \overset{N}{24,000}} \times 100 = 2.0\%$$

$$3.\ \text{Capacity} = \frac{\overset{A}{307.88} \times \overset{L}{3} \times \overset{N}{24,000} \times 1.24675}{\underset{10}{D}}$$

$$= 2,763,716 \text{ gallons.}$$

4. Percentage of total frictions

$$= \left[1 - \frac{\overset{A}{307.88} \times (\overset{P}{95} + \overset{p}{3.69} + \overset{s}{4.33})}{(\overset{A_{s1}}{307.88} \times \overset{M.E.P._1}{59.25}) + (\overset{A_{s2}}{1250.36} \times \overset{M.E.P._2}{13.6})} \right] \times 100$$

$= 9.0\%$.

8. The method to be followed in conducting the boiler test which is recommended is the standard mode determined upon by the Society's Committee on Boiler Trials.* It is suggested that the results of the boiler test be expressed, not only in terms of the number of pounds of water evaporated per pound of coal, in

* Vol. VI., p. 267, *Transactions A. S. M. E.*, 1885.

the customary manner, but also in terms of the number of pounds of coal required to generate 1,000,000 heat units, so that a simple calculation may determine the duty of the engine, if desired, on the basis of 100 lbs. of the coal actually used.

9. In order that the contract between builder and purchaser of a pumping engine may conform to the proposed standard, the guarantee as to performance should be expressed in the following terms:

> 1. The engine shall perform a duty, based upon plunger displacement, equivalent to not less than....foot pounds of work for each one million heat units consumed.
>
> 2. The leakage of the pump shall not exceed....per cent. of the total plunger displacement, when the engine is working at its rated capacity.
>
> 3. The boiler shall supply one million heat units to the engine on a consumption of....pounds of....coal, or it shall evaporate not less than....pounds of water from and at 212 degrees per pound of the combustible portion of the coal named.
>
> 4. The mode of determining these quantities is to conform to the standard method of conducting duty trials recommended by the Committee of the American Society of Mechanical Engineers.

Should one contractor furnish the engine, and another the boiler, separate guarantees will be made, the individual requirements of which are the same as those noted.

It is desirable, where both parties concur therein, to introduce into the contract the following additional provision regarding friction, viz.:

> "The friction of the engine shall not exceed....per cent. of the indicated power developed in the steam-cylinders."

10. Having thus far noted the main principles which have been followed, and having pointed out the various steps by which the results of a test are computed, the committee now beg to submit, in the following pages, the full particulars regarding the standard method of conducting duty trials which they recommend.

The general mode of operation is to first subject the plant to a preliminary run under the working conditions, for the purpose of determining the temperature of the feed-water, or the several temperatures where there is more than one supply. It is usually impracticable to weigh the main supply of water, derived, as it generally is, from a low-placed hot well, and the test of the main quantity of feed-water used must, as a rule, be made with cold water drawn from the service main. The changed conditions in the working of the plant thus introduced, and the arrangement

of apparatus which is frequently needed to measure the additional supplies of feed-water, make it desirable to obtain the working temperatures as a preliminary to the main duty trial. Hence the preliminary run is made, as noted, merely for securing the temperatures. The main test of the boiler and engine is then carried forward, and during this test the weights of the various supplies of feed-water are determined, and the remaining data needed for making the computations. Finally, as soon as practicable after these tests are completed, the rate of leakage through the pump is measured, with the engine at rest.

As to the duration of the test, it appears to the committee that, so far as the main trial is concerned, which is practically a feed-water test, it need not be prolonged more than ten hours, unless, in that time, appreciable errors should be produced by inaccuracies in the observations of the height of water in the gauge glass. The duration of the boiler trial might, with good reason, be made longer, were it not that the results of the boiler test are independent of those of the duty trial. It is desirable to reduce, if possible, the number of hours of the trial to such a point that the time expended upon the work, including that required in preparation for the beginning of the test, and that spent in bringing the test to a close, shall be such that the same expert, without undue physical exertion, may have the test under his continuous supervision from beginning to end. This is feasible where the length of the duty trial, according to the plan proposed, does not exceed ten hours.

Shortly after the committee were appointed several pumping-engine manufacturers were asked to submit their ideas as to the best form of standard to use. The committee take pleasure in acknowledging the receipt of suggestions from the Davidson Steam Pump Company, the Deane Steam Pump Company, the Holly Manufacturing Company, and the George F. Blake Manufacturing Company, the three last named having given full expression to their views upon the subject.

Respectfully submitted,

GEO. H. BARRUS,
A. F. NAGLE,
EDWIN REYNOLDS,
J. J. DE KINDER,
J. S. COON,
Committee.

STANDARD METHOD OF CONDUCTING DUTY TRIALS.

1. TEST OF FEED-WATER TEMPERATURES.

The plant is subjected to a preliminary run, under the conditions determined upon for the test, for a period of three hours, or such a time as is necessary to find the temperature of the feed-water (or the several temperatures, if there is more than one supply) for use in the calculation of the duty. During this test observations of the temperature are made every fifteen minutes. Frequent observations are also made of the speed, length of stroke, indication of water-pressure gauges, and other instruments, so as to have a record of the general conditions under which this test is made.

DIRECTIONS FOR OBTAINING FEED-WATER TEMPERATURES.

When the feed-water is all supplied by one feeding instrument, the temperature to be found is that of the water in the feed-pipe near the point where it enters the boiler. If the water is fed by an injector this temperature is to be corrected for the heat added to the water by the injector, and for this purpose the temperatures of the water entering and of that leaving the injector are both observed. If the water does not pass through a heater on its way to the boiler (that is, that form of heater which depends upon the rejected heat of the engine, such as that contained in the exhaust steam either of the main cylinders or of the auxiliary pumps) it is sufficient, for practical purposes, to take the temperature of the water at the source of supply, whether the feeding instrument is a pump or an injector.

When there are two independent sources of feed-water supply, one the main supply from the hot well, or from some other source, and the other an auxiliary supply derived from the water condensed in the jackets of the main engine and in the live-steam reheater, if one be used, they are to be treated independently. The remarks already made apply to the first, or main, supply. The temperature of the auxiliary supply, if carried by an independent pipe either direct to the boiler or to the main feed-pipe near the boiler, is to be taken at a convenient point in the independent pipe.

When a separator is used in the main steam-pipe, arranged so as to discharge the entrained water back into the boiler by gravity, no account need be made of the temperature of the water thus returned. Should it discharge either into the atmosphere to waste, to the hot well, or to the jacket tank, its temperature is to be determined at the point where the water leaves the separator before its pressure is reduced.

When a separator is used, and it drains by gravity into the jacket tank, this tank being subjected to boiler pressure, the temperatures of the separator water and jacket water are each to be taken before their entrance to the tank.

Should there be any other independent supply of water, the temperature of that is also to be taken on this preliminary test.

DIRECTIONS FOR MEASUREMENT OF FEED-WATER.

As soon as the feed-water temperatures have been obtained the engine is stopped, and the necessary apparatus arranged for determining the weight of the feed-water consumed, or of the various supplies of feed-water, if there is more than one.

In order that the main supply of feed-water may be measured, it will generally be found desirable to draw it from the cold-water service main. The best form of apparatus for weighing the water consists of two tanks, one of which rests upon a platform scale supported by staging, while the other is placed underneath. The water is drawn from the service main into the upper tank, where it is weighed, and it is then emptied into the lower tank. The lower tank serves as a reservoir, and to this the suction-pipe of the feeding apparatus is connected.

The jacket water may be measured by using a pair of small barrels, one being filled while the other is being weighed and emptied. This water, after being measured, may be thrown away, the loss being made up by the main feed-pump. To prevent evaporation from the water, and consequent loss on account of its highly heated condition, each barrel should be partially filled with cold water previous to using it for collecting the jacket water, and the weight of this water treated as tare.

When the jacket water drains back by gravity to the boiler, waste of live steam during the weighing should be prevented by providing a small vertical chamber, and conducting the water into this receptacle before its escape. A glass water-gauge is attached, so as to show the height of water inside the chamber, and this serves as a guide in regulating the discharge valve. The chamber may be made of piping in the manner shown in the appended figure (108).

Fig. 108

When the jacket water is returned to the boiler by means of a pump, the discharge valve should be throttled during the test, so that the pump may work against its usual pressure—that is, the boiler pressure as nearly as may be, a gauge being attached to the discharge pipe for this purpose.

When a separator is used and the entrained water discharges either to waste, to the hot well, or to the jacket tank, the weight of this water is to be determined, the water being drawn into barrels in the manner pointed out for measuring the jacket water. Except in the case where the separator discharges into the jacket tank, the entrained water thus found is treated, in the calculations, in the same manner as moisture shown by

the calorimeter test. When it discharges into the jacket tank, its weight is simply subtracted from the total weight of water fed, and allowance made for heat of this water lost by radiation between separator and tank.

When the jackets are drained by a trap, and the condensed water goes either to waste or to the hot well, the determination of the quantity used is not necessary to the main object of the duty trial, because the main feed-pump in such cases supplies all the feed-water. For the sake of having complete data, however, it is desirable that this water be measured, whatever the use to which it is applied.

Should live steam be used for reheating the steam in the intermediate receiver, it is desirable to separate this from the jacket steam, if it drains into the same tank, and measure it independently. This, likewise, is not essential to the main object of the duty trial, though useful for purposes of information.

The remarks as to the manner of preventing losses of live steam and of evaporation, in the measurement of jacket water, apply to the measurement of any other hot water under pressure which may be used for feed-water.

Should there be any other independent supply of water to the boiler besides those named, its quantity is to be determined independently, apparatus for all these measurements being set up during the interval between the preliminary run and the main trial, when the plant is idle.

2. THE MAIN DUTY TRIAL.

The duty trial is here assumed to apply to a complete plant, embracing a test of the performance of the boiler, as well as that of the engine. The test of the two will go on simultaneously after both are started, but the boiler test will begin a short time in advance of the commencement of the engine test, and continue a short time after the engine test is finished. The mode of procedure is as follows:

The plant having been worked for a suitable time under normal conditions, the fire is burned down to a low point and the engine brought to rest. The fire remaining on the grate is then quickly hauled, the furnace cleaned, and the refuse withdrawn from the ash pit. The boiler test is now started, and this test is made in accordance with the rules for a standard method recommended by the Committee on Boiler Tests of the American Society of Mechanical Engineers.* This method, briefly described, consists in starting the test with a new fire lighted with wood, the boiler having previously been heated to its normal working degree; operating the boiler in accordance with the conditions determined upon; weighing coal, ashes, and feed-water; observing the draught, temperatures of feed-water and escaping gases, and such other data as may be incidentally desired; determining the quantity of moisture in the coal and in the steam; and at the close of

* Vol. VI., p. 267, *Transactions A. S. M. E.*, 1885.

the test hauling the fire, and deducting from the weight of coal fired whatever unburned coal is contained in the refuse withdrawn from the furnace, the quantity of water in the boiler and the steam pressure being the same as at the time of lighting the fire at the beginning of the test.

Previous to the close of the test it is desirable that the fire should be burned down to a low point, so that the unburned coal withdrawn may be in a nearly consumed state. The temperature of the feed-water is observed at the point where the water leaves the engine heater, if this be used, or at the point where it enters the flue heater, if that apparatus be employed. Where an injector is used for supplying the water, a deduction is to be made in either case for the increased temperature of the water derived from the steam which it consumes.

As soon after the beginning of the boiler test as practicable the engine is started and preparations are made for the beginning of the engine test. The formal commencement of this test is delayed till the plant is again in normal working condition, which should not be over one hour after the time of lighting the fire. When the time for commencement arrives the feed-water is momentarily shut off, and the water in the lower tank is brought to a mark. Observations are then made of the number of tanks of water thus far supplied, the height of water in the gauge-glass of the boiler, the indication of the counter on the engine, and the time of day; after which the supply of feed-water is renewed, and the regular observations of the test, including the measurement of the auxiliary supplies of feed-water, are commenced. The engine test is to continue at least ten hours. At its expiration the feed-pump is again momentarily stopped, care having been taken to have the water slightly higher than at the start, and the water in the lower tank is brought to the mark. When the water in the gauge-glass has settled to the point which it occupied at the beginning, the time of day and the indication of the counter are observed, together with the number of tanks of water thus far supplied, and the engine test is held to be finished. The engine continues to run after this time till the fire reaches a condition for hauling, and completing the boiler test. It is then stopped, and the final observations relating to the boiler test are taken.

The observations to be made and data obtained for the purposes of the engine test, or duty trial proper, embrace the weight of feed-water supplied by the main feeding apparatus, that of the water drained from the jackets, and any other water which is ordinarily supplied to the boiler, determined in the manner pointed out. They also embrace the number of hours duration, and number of single strokes of the pump during the test; and, in direct-acting engines, the length of the stroke; together with the indications of the gauges attached to the force and suction mains, and indicator diagrams from the steam-cylinders. It is desirable that pump diagrams also be obtained.

Observations of the length of stroke, in the case of direct-acting engines, should be made every five minutes; observations of the water-pressure gauges every fifteen minutes; observations of the remaining instruments, such as steam-gauge, vacuum-gauge, thermometer in pump well, thermometer in feed-pipe, thermometer showing tempera-

ture of engine room, boiler room, and outside air, thermometer in flue, thermometer in steam-pipe, if the boiler has steam-heating surface, barometer, and other instruments which may be used, every half-hour. Indicator diagrams should be taken every half-hour.

When the duty trial embraces simply a test of the engine, apart from the boiler, the course of procedure will be the same as that described, excepting that the fires will not be hauled, and the special observations relating to the performance of the boiler will not be taken.

DIRECTIONS REGARDING ARRANGEMENT AND USE OF INSTRUMENTS, AND OTHER PROVISIONS FOR THE TEST.

The gauge attached to the force main is liable to a considerable amount of fluctuation unless the gauge-cock is nearly closed. The practice of choking the cock is objectionable. The difficulty may be satisfactorily overcome, and a nearly steady indication secured, with cock wide open, if a small reservoir having an air-chamber is interposed between the gauge and the force main, in the manner shown in the appended figure (109). By means of a gauge-glass on the side of the chamber and an air-valve, the average water-level may be adjusted to the height of the centre of the gauge, and correction for this element of variation is avoided. If not thus adjusted, the reading is to be referred to the level shown, whatever this may be.

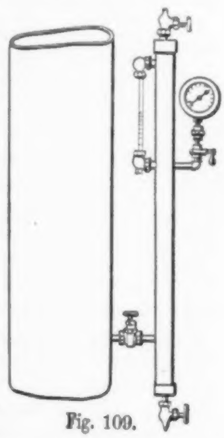

Fig. 109.

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water-cylinders, in a position parallel to the piston-rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed, and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, such a method of measurement is preferred. The personal errors in observing the exact scale marks, which are liable to creep in, may thereby be avoided.

The form of calorimeter to be used for testing the quality of the steam is left to the decision of the person who conducts the trial. It is preferred that some form of continuous calorimeter be used, which acts directly on the moisture tested. If either the superheating calorimeter * or the wire-drawing † instrument be employed, the steam which

* Vol. VII., p. 178, 1886.

† Vol. XI., p. 790, 1890, *Transactions A. S. M. E.* (Paper on "A Universal Calorimeter," May, 1890.)

it discharges is to be measured either by numerous short trials, made by condensing it in a barrel of water previously weighed, thereby obtaining the rate by which it is discharged, or by passing it through a surface condenser of some simple construction, and measuring the whole quantity consumed. When neither of these instruments is at hand, and dependence must be placed upon the barrel calorimeter, scales should be used which are sensitive to a change in weight of a small fraction of a pound, and thermometers which may be read to tenths of a degree. The pipe which supplies the calorimeter should be thoroughly warmed and drained just previous to each test. In making the calculations the specific heat of the material of the barrel or tank should be taken into account, whether this be of metal or of wood.

If the steam is superheated, or if the boiler is provided with steam-heating surface, the temperature of the steam is to be taken by means of a high-grade thermometer resting in a cup holding oil or mercury, which is screwed into the steam-pipe so as to be surrounded by the current of steam. The temperature of the feed-water is preferably taken by means of a cup screwed into the feed-pipe in the same manner.

Indicator pipes and connections used for the water-cylinders should be of ample size, and, so far as possible, free from bends. Three-quarter-inch pipes are preferred, and the indicators should be attached one at each end of the cylinder. It should be remembered that indicator springs which are correct under steam heat are erroneous when used for cold water. When such springs are used, the actual scale should be determined, if calculations are made of the indicated work done in the water-cylinders. The scale of steam-springs should be determined by a comparison, under steam pressure, with an accurate steam-gauge at the time of the trial, and that of water-springs by cold dead-weight test.

The accuracy of all the gauges should be carefully verified by comparison with a reliable mercury column. Similar verification should be made of the thermometers, and if no standard is at hand, they should be tested in boiling water and melting ice.

To avoid errors in conducting the test, due to leakage of stop-valves either on the steam-pipes, feed-water pipes, or blow-off pipes, all these pipes not concerned in the operation of the plant under test should be disconnected.

3. LEAKAGE TEST OF PUMP.

As soon as practicable after the completion of the main trial (or at some time immediately preceding the trial), the engine is brought to rest, and the rate determined at which leakage takes place through the plunger and valves of the pump, when these are subjected to the full pressure of the force main.

The leakage of the plunger is most satisfactorily determined by making the test with the cylinder head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception

of an overflow-pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and the water from the force main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured.

Should the escape of the water into the engine room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow pipe in the wooden head. The apparatus may be constructed, if desired, in a somewhat rude manner, and yet be sufficiently accurate for practical requirements. The test should be made, if possible, with the plunger in various positions.

In the case of a pump so planned that it is difficult to remove the cylinder head, it may be desirable to take the leakage from one of the openings which are provided for the inspection of the suction valves, the head being allowed to remain in place.

It is here assumed that there is a practical absence of valve leakage, a condition of things which ought to be attained in all well-constructed pumps. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger test. Leakage of the discharge valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction valves will be shown by the disappearance of water which covers them.

If valve leakage is found which cannot be remedied, the quantity of water thus lost should also be tested. The determination of the quantity which leaks through the suction valves, where there is no gate in the suction pipe, must be made by indirect means. One method is to measure the amount of water required to maintain a certain pressure in the pump cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge valves of the pump.

The exact methods to be followed in any particular case, in determining leakage, must be left to the judgment and ingenuity of the person conducting the test.

4. TABLE OF DATA AND RESULTS.

In order that uniformity may be secured, it is suggested that the data and results, worked out in accordance with the standard method, be tabulated in the manner indicated in the following scheme :

DUTY TRIAL OF ENGINE.

DIMENSIONS.

1. Number of steam-cylinders ....
2. Diameter of steam-cylinders.............................. ins.
3. Diameter of piston-rods of steam-cylinders................ ins.

4. Nominal stroke of steam-pistons........................ ft.
5. Number of water-plungers..............................
6. Diameter of plungers.................................. ins.
7. Diameter of piston-rods of water-cylinders............ ins.
8. Nominal stroke of plungers............................ ft.
9. Net area of plungers.................................. sq. ins.
10. Net area of steam-pistons............................ sq. ins.
11. Average length of stroke of steam-pistons during trial..... ft.
12. Average length of stroke of plungers during trial......... ft.

(Give also complete description of plant.)

TEMPERATURES.

13. Temperature of water in pump well........................ degs.
14. Temperature of water supplied to boiler by main feed-pump. degs.
15. Temperature of water supplied to boiler from various other sources .... degs.

FEED-WATER.

16. Weight of water supplied to boiler by main feed-pump.... lbs.
17. Weight of water supplied to boiler from various other sources.. lbs.
18. Total weight of feed-water supplied from all sources...... lbs.

PRESSURES.

19. Boiler pressure indicated by gauge...................... lbs.
20. Pressure indicated by gauge on force main................ lbs.
21. Vacuum indicated by gauge on suction main............ ins.
22. Pressure corresponding to vacuum given in preceding line . lbs.
23. Vertical distance between the centres of the two gauges... ins.
24. Pressure equivalent to distance between the two gauges... lbs.

MISCELLANEOUS DATA.

25. Duration of trial.. hrs.
26. Total number of single strokes during trial................
27. Percentage of moisture in steam supplied to engine, or number of degrees of superheating......................... % or deg.
28. Total leakage of pump during trial, determined from results of leakage test.................................. lbs.
29. Mean effective pressure, measured from diagrams taken from steam-cylinders................................. M. E. P.

PRINCIPAL RESULTS.

30. Duty .... ft. lbs.
31. Percentage of leakage.................................. %
32. Capacity .... gals.
33. Percentage of total frictions.......................... %

ADDITIONAL RESULTS.*

34. Number of double strokes of steam-piston per minute .....
35. Indicated horse-power developed by the various steam-cylinders.. I. H. P.

* These are not necessary to the main object, but it is desirable to give them.

36. Feed-water consumed by the plant per hour .... lbs.
37. Feed-water consumed by the plant per indicated horse-power per hour, corrected for moisture in steam......... lbs.
38. Number of heat units consumed per indicated horse-power per hour...... .... B. T. U.
39. Number of heat units consumed per indicated horse-power per minute .... B. T. U.
40. Steam accounted for by indicator at cut-off and release in the various steam-cylinders............................ lbs.
41. Proportion which steam accounted for by indicator bears to the feed-water consumption..........................

SAMPLE DIAGRAMS TAKEN FROM STEAM-CYLINDERS.

[Also, if possible, full measurements of the diagrams, embracing pressures at the initial point, cut-off, release, and compression; also back-pressure, and the proportions of the stroke completed at the various points noted.]

42. Number of double strokes of pump per minute............
43. Mean effective pressure, measured from pump diagrams... M. E. P.
44. Indicated horse-power exerted in pump cylinders.......... I. H. P.

SAMPLE DIAGRAMS TAKEN FROM PUMP CYLINDERS.

DATA AND RESULTS OF BOILER TEST.

[IN ACCORDANCE WITH THE SCHEME RECOMMENDED BY THE BOILER TEST COMMITTEE OF THE SOCIETY.]

1. Date of trial..
2. Duration of trial...................................... hrs.

DIMENSIONS AND PROPORTIONS.

3. Grate surface wide long Area........... sq. ft.
4. Water-heating surface. .... sq. ft.
5. Superheating surface................................... sq. ft.
6. Ratio of water-heating surface to grate surface............

(Give also complete description of boilers.)

AVERAGE PRESSURES.

7. Steam pressure in boiler by gauge....................... lbs.
8. Atmospheric pressure by barometer...................... lbs.
9. Force of draught in inches of water..................... ins.

AVERAGE TEMPERATURES.

10. Of steam.. degs.
11. Of escaping gases................................. degs.
12. Of feed-water..................................... degs.

FUEL.

13. Total amount of coal consumed * .... lbs.
14. Moisture in coal .... %
15. Dry coal consumed .... lbs.
16. Total refuse (dry) .... lbs.
17. Total combustible (dry weight of coal, item 15, less refuse, item 16) .... lbs.
18. Dry coal consumed per hour .... lbs.

RESULTS OF CALORIMETRIC TEST.

19. Quality of steam, dry steam being taken as unity ....
20. Percentage of moisture in steam .... %
21. Number of degrees superheated .... degs.

WATER.

22. Total weight of water pumped into boiler and apparently evaporated † .... lbs.
23. Water actually evaporated corrected for quality of steam .. lbs.
24. Equivalent water evaporated into dry steam from and at 212° Fahr. ‡ .... lbs.
25. Equivalent total heat derived from fuel, in British Thermal Units .... B. T. U.
26. Equivalent water evaporated into dry steam from and at 212° Fahr. per hour .... lbs.

ECONOMIC EVAPORATION.

27. Water actually evaporated per pound of dry coal from actual pressure and temperature .... lbs.
28. Equivalent water evaporated per pound of dry coal from and at 212° Fahr .... lbs.
29. Equivalent water evaporated per pound of combustible from and at 212° Fahr .... lbs.
30. Number of pounds of coal required to supply 1,000,000 British Thermal Units .... lbs.

RATE OF COMBUSTION.

31. Dry coal actually burned per square foot of grate surface per hour .... lbs.

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* Including equivalent of wood used in lighting fire. One pound of wood equals 0.4 of a pound of coal, not including unburned coal withdrawn from fire at end of test.

† Corrected for inequality of water-level and of steam pressure at beginning and end of test.

‡ Factor of evaporation $= \frac{H - h}{965.7}$, H and h being respectively the total heat units in steam of the average observed pressure, and in water of the average observed temperature of feed.

RATE OF EVAPORATION.

32. Water evaporated from and at 212° Fahr. per square foot of heating surface per hour .... lbs.

[NOTE.—To determine the percentage of surface moisture in the coal a sample of the coal should be dried for a period of twenty-four hours, being subjected to a temperature of not more than 212°. The quantity of unconsumed coal contained in the refuse withdrawn from the furnace and ash pit at the end of the test may be found by sifting either the whole of the refuse, or a sample of the same, in a screen having $\frac{3}{8}$-inch meshes. This, deducted from the weight of dry coal fired, gives the weight of dry coal consumed, for line 15.—*Duty Trial Committee.*]

APPENDIX.

MEASUREMENT OF WATER BY MEANS OF WEIRS, TUBES, AND NOZZLES.

THE following memoranda in regard to the measurement of water by means of weirs, tubes, and nozzles are appended to the report, in order that these systems may be readily availed of whenever it is practicable to do so. It is not attempted to give full directions here, but simply the general principles which should be followed in order to obtain reliable work. The reader is referred to the accounts of various investigators themselves, who have experimented in these lines, for the detailed instructions which cannot here be introduced:

WEIRS.

The measurement of water by the use of weirs is generally based, at the present time, on the results of experiments made by Mr. James B. Francis, C.E., at Lowell, in 1852, an account of which is given in *Lowell Hydraulic Experiments*, D. Van Nostrand. These experiments led to the construction of the following formula, which is known as the "Francis Formula," viz.:

$$Q = 3.33\,(L - 0.2H) \times H^{\frac{3}{2}},$$

in which Q is the discharge of water in cubic feet per second, L the length of the weir, and H the depth on the weir, all of these measurements being in terms of the English foot. The coefficient, 3.33, was obtained from the mean of eighty-eight experiments, the greatest variation from the mean in any individual case being one per cent. The length of the weir, in all but six of the experiments, was approximately ten feet. The depth of water on the weir varied from 7 to 19 inches. The formula applies to that type of weir having perfect contraction at each end, which was the form used in sixty-five experiments.

The weir was of rectangular cross-section, with a horizontal crest and vertical ends. The upper edge was made of cast iron, and the corner presented to the current was square and sharp. The horizontal part of the crest was one-fourth of an inch wide, and the remaining part was bevelled off at an angle of 45°. The ends were of similar cross-section to the crest. The depth on the weir was taken by means of hook gauges, six feet from the weir, these gauges being placed in wooden boxes situated on the sides of the canal and communicating with the water through small openings. Vertical gratings, for overcoming eddies in the current, were provided above the gauges. The distance from the side of the canal to the end of the weir was about two feet, and the depth of the canal below the crest was, in most of the experiments, about five feet.

The Francis Formula is applicable only to cases similar to the ones described. According to Mr. Francis' statement, it cannot be applied where the depth on the weir exceeds one-third of the length, nor to very small depths: furthermore, the

distance from the side of the canal to the end of the weir should not be less than three times that on the weir.

In using the formula, the depth of water on the weir should be corrected for the head due to the velocity of approach according to the formula

$$H' = (H + h^{\frac{3}{2}} - h^{\frac{3}{2}})^{\frac{2}{3}},$$

in which H' is the corrected depth, H the observed depth, and h the head due to the velocity of approach. This last may be determined from the formula

$$h = \frac{V}{64.3},$$

in which V is the velocity of approach in feet per second, which may be determined by dividing the uncorrected discharge of water, in cubic feet per second, by the area of the cross-section of the stream flowing through the canal, in square feet.

The reader is also referred to the experiments on weirs made by Hamilton Smith, Jr., described in *Smith's Hydraulics*, and to those made by Fteley and Stearns, described in *Transactions American Society of Civil Engineers*, 1883.

VENTURI TUBES.

Mr. Clemens Herschel, C.E., Holyoke, Mass., in a paper read before the American Society of Civil Engineers, December 21, 1887, and printed in their *Transactions*, November, 1887, and January, 1888, recommends the use of a Venturi tube, inserted in the force main of the pumping engine, for determining the quantity of water discharged. Such a tube, applied, for example, to a 24-inch main, has a total length of about twenty feet. At a distance of four feet from the end nearest the engine, the inside diameter of the tube is contracted to a throat having a diameter of about eight inches. A pressure gauge is attached to each of two chambers, the one surrounding and communicating with the entrance, or main pipe, the other with the throat. According to experiments which Mr. Herschel has made upon two tubes of this kind, one of which was four inches in diameter at the throat and twelve inches at its entrance, and the other about thirty-six inches in diameter at the throat and nine feet at its entrance, the quantity of water which passes through the tube is very nearly the theoretical discharge through an opening having an area equal to that of the throat, and a velocity which is that due to the difference in head shown by the two gauges. Mr. Herschel states that the coefficient for these two widely varying sizes of tubes and for a wide range of velocity through the pipe was found to be within two per cent., either way, of 98%. In other words, the quantity of water flowing through the tube per second is expressed within two per cent. by the formula

$$W = 0.98 \times A \times \sqrt{2gh},$$

in which A is the area of the throat of the tube, and h the head, in feet, corresponding to the difference in the pressure of the water entering the tube and that found at the throat.

The accuracy of this form of measurement has not been determined under conditions which apply exactly to pumping engines, but the results here alluded to give promise of its becoming a valuable aid for this purpose, when thoroughly developed.

NOZZLES.

The measurement of water by computation from its discharge through orifices, or through the nozzles of fire-hose, furnishes a means of determining the quantity of water delivered by a pumping engine which can be applied without much difficulty. Mr. John R. Freeman, C.E., of Boston, has carried on a series of investigations upon fire-nozzles, described in the *Transactions American Society of Civil Engineers*, November, 1889, which are of value in this connection. Mr. Freeman's experiments covered a wide range of pressures and sizes, and the results showed that the coefficient of discharge, for a smooth nozzle of ordinary good form, was within one-half of one per cent., either way, of 0.977; the diameter of the nozzle being accurately calipered, and the pressures being determined by means of an accurate gauge attached to a suitable piezometer at the base of the play-pipe.

In order to use this method for determining the quantity of water discharged by a pumping engine, it would be necessary to provide a pressure box, to which the water would be conducted, and attach to the box as many nozzles as would be required to carry off the water. According to Mr. Freeman's estimate, four 1¼-inch nozzles, thus connected, with a pressure of 80 lbs. per square inch, would discharge the full capacity of a two-and-a-half-million engine. To serve the same end, Mr. Freeman suggests, in the *Journal of the New England Water Works Association*, March, 1890, the use of a portable apparatus with a single opening for discharge, consisting essentially of a Siamese nozzle, so called, the water being carried to it by three or more lines of fire-hose.

To insure reliability for these measurements, it is necessary that the shut-off valve in the force main, or the several shut-off valves, should be tight, so that all the water discharged by the engine may pass through the nozzles.

TABLE No. 1.

PROPERTIES OF SATURATED STEAM.

[From Charles T. Porter's treatise on *The Richards' Steam-Engine Indicator.*]

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
1	102.00	102.08	1042.96	1145.05	.0030
2	126.26	126.44	1026.01	1152.45	.0058
3	141.62	141.87	1015.25	1157.13	.0085
4	153.07	153.39	1007.22	1160.62	.0112
5	162.33	162.72	1000.72	1163.44	.0137
6	170.12	170.57	995.24	1165.82	.0163
7	176.91	177.42	990.47	1167.89	.0189
8	182.91	183.48	986.24	1169.72	.0214
9	188.31	188.94	982.43	1171.37	.0239
10	193.24	193.91	978.95	1172.87	.0264
11	197.76	198.49	975.76	1174.25	.0289
12	201.96	202.73	972.80	1175.53	.0313
13	205.88	206.70	970.02	1176.73	.0337
14	209.56	210.42	967.42	1177.85	.0362
15	213.02	213.93	964.97	1178.91	.0387
16	216.29	217.25	962.65	1179.90	.0413
17	219.41	220.40	960.45	1180.85	.0437
18	222.37	223.41	958.34	1181.76	.0462
19	225.20	226.28	956.34	1182.62	.0487
20	227.91	229.03	954.41	1183.45	.0511
21	230.51	231.67	952.57	1184.24	.0536
22	233.01	234.21	950.79	1185.00	.0561
23	235.43	236.67	949.07	1185.74	.0585
24	237.75	239.02	947.42	1186.45	.0610
25	240.00	241.31	945.82	1187.13	.0634
26	242.17	243.52	944.27	1187.80	.0658
27	244.28	245.67	942.77	1188.44	.0683
28	246.32	247.74	941.32	1189.06	.0707
29	248.31	249.76	939.90	1189.67	.0731
30	250.24	251.73	938.92	1190.26	.0755
31	252.12	253.64	937.18	1190.83	.0779
32	253.95	255.51	935.88	1191.39	.0803
33	255.73	257.32	934.60	1191.93	.0827
34	257.47	259.10	933.36	1192.46	.0851
35	259.17	260.83	932.15	1192.98	.0875
36	260.83	262.52	930.96	1193.49	.0899
37	262.45	264.18	929.80	1193.98	.0922
38	264.04	265.80	928.67	1194.47	.0946
39	265.59	267.38	927.56	1194.94	.0970
40	267.12	268.93	926.47	1195.41	.0994
41	268.61	270.46	925.40	1195.86	.1017
42	270.07	271.95	924.35	1196.31	.1041
43	271.50	273.41	923.33	1196.74	.1064
44	272.91	274.85	922.32	1197.17	.1088
45	274.29	276.26	921.33	1197.60	.1111
46	275.65	277.65	920.36	1198.01	.1134

TABLE No. 1.—*Continued.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
47	276.98	279.01	919.40	1198.42	.1158
48	278.29	280.35	918.46	1198.82	.1181
49	279.58	281.67	917.54	1199.21	.1204
50	280.85	282.96	916.63	1199.60	.1227
51	282.09	284.24	915.73	1199.98	.1251
52	283.32	285.49	914.85	1200.35	.1274
53	284.53	286.73	913.98	1200.72	.1297
54	285.72	287.95	913.13	1201.08	.1320
55	286.89	289.15	912.29	1201.44	.1343
56	288.05	290.33	911.46	1201.79	.1366
57	289.11	291.50	910.64	1202.14	.1388
58	290.31	292.65	909.83	1202.48	.1411
59	291.42	293.79	909.03	1202.82	.1434
60	292.52	294.91	908.24	1203.15	.1457
61	293.59	296.01	907.47	1203.48	.1479
62	294.66	297.10	906.70	1203.81	.1502
63	295.71	298.18	905.94	1204.13	.1524
64	296.75	299.24	905.20	1204.44	.1547
65	297.77	300.30	904.46	1204.76	.1569
66	298.78	301.33	903.73	1205.07	.1592
67	299.78	302.36	903.01	1205.37	.1614
68	300.77	303.37	902.29	1205.67	.1637
69	301.75	304.38	901.59	1205.97	.1659
70	302.71	305.37	900.89	1206.26	.1681
71	303.67	306.35	900.21	1206.56	.1703
72	304.61	307.32	899.52	1206.84	.1725
73	305.55	308.27	898.85	1207.13	.1748
74	306.47	309.22	898.18	1207.41	.1770
75	307.38	310.16	897.52	1207.69	.1792
76	308.29	311.09	896.87	1207.96	.1814
77	309.18	312.01	896.23	1208.24	.1836
78	310.06	312.92	895.59	1208.51	.1857
79	310.94	313.82	894.95	1208.77	.1879
80	311.81	314.71	894.33	1209.04	.1901
81	312.67	315.59	893.70	1209.30	.1923
82	313.52	316.46	893.09	1209.56	.1945
83	314.36	317.33	892.48	1209.82	.1967
84	315.19	318.19	891.88	1210.07	.1988
85	316.02	319.04	891.28	1210.32	.2010
86	316.83	319.88	890.69	1210.57	.2032
87	317.65	320.71	890.10	1210.82	.2053
88	318.45	321.54	889.52	1211.06	.2075
89	319.24	322.36	888.94	1211.31	.2097
90	320.03	323.17	888.37	1211.55	.2118
91	320.82	323.98	887.80	1211.79	.2139
92	321.59	324.78	887.24	1212.02	.2160
93	322.36	325.57	886.68	1212.26	.2182
94	323.12	326.35	886.13	1212.49	.2204
95	323.88	327.13	885.58	1212.72	.2224

TABLE No. 1.—*Continued.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
96	324.63	327.90	885.04	1212.95	.2245
97	325.37	328.67	884.50	1213.18	.2266
98	326.11	329.43	883.97	1213.40	.2288
99	326.84	330.18	883.44	1213.62	.2309
100	327.57	330.93	882.91	1213.84	.2330
101	328.29	331.67	882.39	1214.06	.2351
102	329.00	332.41	881.87	1214.28	.2371
103	329.71	333.14	881.35	1214.50	.2392
104	330.41	333.86	880.84	1214.71	.2413
105	331.11	334.58	880.34	1214.92	.2434
106	331.80	335.30	879.84	1215.14	.2454
107	332.49	336.00	879.34	1215.35	.2475
108	333.17	336.71	878.84	1215.55	.2496
109	333.85	337.41	878.35	1215.76	.2516
110	334.52	338.10	877.86	1215.97	.2537
111	335.19	338.79	877.37	1216.17	.2558
112	335.85	339.47	876.89	1216.37	.2578
113	336.51	340.15	876.41	1216.57	.2599
114	337.16	340.83	875.94	1216.77	.2619
115	337.81	341.50	875.47	1216.97	.2640
116	338.45	342.16	875.00	1217.17	.2661
117	339.10	342.83	874.53	1217.36	.2681
118	339.73	343.48	874.07	1217.56	.2702
119	340.36	344.14	873.61	1217.75	.2722
120	340.99	344.78	873.15	1217.94	.2742
121	341.61	345.43	872.70	1218.13	.2762
122	342.23	346.07	872.25	1218.32	.2782
123	342.85	346.70	871.80	1218.51	.2802
124	343.46	347.34	871.35	1218.69	.2822
125	344.07	347.97	870.91	1218.88	.2842
126	344.67	348.59	870.47	1219.06	.2862
127	345.27	349.21	870.03	1219.25	.2882
128	345.87	349.83	869.59	1219.43	.2902
129	346.45	350.44	869.16	1219.61	.2922
130	347.05	351.05	868.73	1219.79	.2942
131	347.64	351.66	868.30	1219.97	.2961
132	348.22	352.26	867.88	1220.15	.2981
133	348.80	352.86	867.46	1220.32	.3001
134	349.38	353.46	867.03	1220.50	.3020
135	349.95	354.05	866.62	1220.67	.3040
136	350.52	354.64	866.20	1220.85	.3060
137	351.08	355.23	866.79	1221.02	.3079
138	351.75	355.81	865.38	1221.19	.3099
139	352.21	356.39	864.97	1221.36	.3118
140	352.76	356.96	864.56	1221.53	.3138
141	353.31	357.54	864.16	1221.70	.3158
142	353.86	358.11	863.76	1221.87	.3178
143	354.41	358.67	863.36	1222.03	.3199
144	354.96	359.24	862.96	1222.20	.3219

TABLE No. 1.—*Continued.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
145	355.50	359.80	862.56	1222.36	.3239
146	356.03	360.85	862.17	1222.53	.3259
147	356.57	360.91	861.78	1222.69	.3279
148	357.10	361.46	861.39	1222.85	.3299
149	357.63	362.01	861.00	1223.01	.3319
150	358.16	362.55	860.62	1223.18	.3340
151	358.68	363.10	860.23	1223.33	.3358
152	359.20	363 64	859.85	1223.49	.3376
153	359.72	364.17	859.47	1223.65	.3394
154	360.23	364.71	859.10	1223.81	.3412
155	360.74	365.24	858.72	1223.97	.3430
156	361.26	365.77	858.35	1224.12	.3448
157	361.76	366.30	857.98	1224.28	.3466
158	362.27	366.82	857.61	1224.43	.3484
159	362.77	367.34	857.24	1224.58	.3502
160	363.27	367.86	856.87	1224.74	.3520
161	363.77	368.38	856.50	1224.89	.3539
162	364.27	368.89	856.14	1225.04	.3558
163	364.76	369.41	855.78	1225.19	.3577
164	365.25	369.92	855.42	1225.34	.3596
165	365.74	370 42	855.06	1225.49	.3614
166	366.23	370.93	854.70	1225.64	.3633
167	366.71	371.43	854.35	1225.78	.3652
168	367.19	371.93	853.99	1225.93	.3671
169	367.68	372.43	853.64	1226.08	.3690
170	368.15	372.93	853.29	1226.22	.3709
171	368.63	373.42	852.94	1226.37	.3727
172	369.10	373.91	852.59	1226.51	.3745
173	369.57	374.40	852.25	1226.66	.3763
174	370.04	374.89	851.90	1226.80	.3781
175	370.51	375.38	851.56	1226.94	.3799
176	370.97	375.86	851.22	1227.08	.3817
177	371.44	376.34	850.88	1227.23	.3835
178	371.90	376.82	850.54	1227.37	.3853
179	372.36	377.30	850.20	1227.51	.3871
180	372.82	377.78	849.86	1227.65	.3889
181	373.27	378.25	849.53	1227.78	.3907
182	373.73	378.72	849.20	1227.92	.3925
183	374.18	379.19	848.86	1228.06	.3944
184	374.63	379.66	848.53	1228.20	.3962
185	375.08	380.13	848.20	1228.33	.3980
186	375.52	380.59	847.88	1228.47	.3999
187	375.97	381.05	847.55	1228.61	.4017
188	376.41	381.51	847.22	1228.74	.4035
189	376 85	381.97	846.90	1228.87	.4053
190	377.29	382.42	846.58	1229.01	.4072
191	377.72	382.88	846.26	1229.14	.4089
192	378.16	383.33	845.94	1229.27	.4107
193	378.59	383.78	845.62	1229.41	.4125

TABLE No. 1.—*Concluded.*

Pressure above zero.	Temperature.	Sensible Heat above zero Fahr.	Latent Heat.	Total Heat above zero Fahr.	Weight of One Cubic Foot.
Lbs. per sq. in.	Fahr. Deg.	B.T.U.	B.T.U.	B.T.U.	Lbs.
194	379.02	384.23	845.30	1229.54	.4143
195	379.45	384.67	844.99	1229.67	.4160
196	379.97	385.12	844.68	1229.80	.4178
197	380.30	385.56	844.36	1229 93	.4196
198	380.72	386.00	844.05	1230.06	.4214
199	381.15	386.44	843 74	1230.19	.4231
200	381.57	386.88	843.43	1230.31	.4249
201	381.99	387.32	843.12	1230.44	.4266
202	382.41	387.76	842.81	1230.57	.4283
203	382.82	388.19	842.50	1230.70	.4300
204	383.24	388.62	842.20	1230 82	.4318
205	383.65	389.05	841.89	1230.95	.4335
206	384.06	389.48	841.59	1231.07	.4352
207	384.47	389.91	841.29	1231.20	.4369
208	384.88	390.33	840.99	1231.32	.4386
209	385.28	390.75	840.69	1231.45	.4403
210	385.67	391.17	840.39	1231.57	.4421

TABLE No. 2.

QUANTITIES OF HEAT CONTAINED IN ONE POUND OF WATER AT VARIOUS TEMPERATURES, RECKONED FROM ZERO, FAHRENHEIT.

[From Charles T. Porter's treatise on *The Richards' Steam-Engine Indicator.*]

Temperature.	Heat contained above zero.	Temperature.	Heat contained above zero.	Temperature.	Heat contained above zero.
Fahr. Deg.	B.T.U.	Fahr. Deg.	B.T.U.	Fahr. Deg.	B.T.U.
35	35.00	155	155.33	275	276.98
40	40.00	160	160.37	280	282.09
45	45.00	165	165.41	285	287.21
50	50.00	170	170.45	290	292.32
55	55.00	175	175.49	295	297.45
60	60.00	180	180.54	300	302.58
65	65.01	185	185.59	305	307.71
70	70.02	190	190.64	310	312.84
75	75.02	195	195.69	315	317.98
80	80.03	200	200.75	320	323.13
85	85.04	205	205.81	325	328.28
90	90.05	210	210.87	330	333.43
95	95.06	215	215.93	335	338.59
100	100.08	220	221.00	340	343.75
105	105.09	225	226.07	345	348.92
110	110.11	230	231.15	350	354.10
115	115.12	235	236.23	355	359.28
120	120.14	240	241 31	360	364.46
125	125.16	245	246.39	365	369.65
130	130.19	250	251.48	370	374.84
135	135.21	255	256.57	375	380.04
140	140.24	260	261.67	380	385.24
145	145.27	265	266.77	385	390.45
150	150.30	270	271.87	390	395.67

NOTE.

This report, presented as one of the papers of the Cincinnati Convention of the Society in May, 1890, is printed here for circulation among those interested and with a view to elicit discussion and suggestion. When such criticism has been received and considered, the report, with appended discussion, will be presented for acceptance as the final views of the Committee, to receive the usual action of the Society upon reports of this kind.

Secretary.

CCLXXXII.

HIRN AND DWELSHAUVERS' THEORY OF THE STEAM-ENGINE, EXPERIMENTAL AND ANALYTIC.

BY R. H. THURSTON, ITHACA, N. Y.

(Member of the Society and Past President.)

WHILE those great mathematical physicists, Rankine and Clausius, were, each in his own way and independently, developing the modern theory of the steam-engine and creating the new science of thermo-dynamics, engineers who were less familiar with the philosophy of the subject on that side, but who were thoroughly familiar with it as practitioners and in its practical operation, were seeking to ascertain the facts and those numerical constants which are required in the construction of an applied science of heat utilization in such machines.

Daniel Kinnear Clark and the two great pioneers in modern thermo dynamic science were working contemporaneously—the one to ascertain just where and how heat was utilized and wasted; the others to ascertain just what were the laws of heat-energy transformations, and the theoretical efficiency of heat-engines. This work began, with all three, previous to 1850.* Before the new science had become recognized and had been given a permanent place and form, M. G. A. Hirn took up the work inaugurated by Clark (1855 and later), and was engaged for many years thereafter in the prosecution of researches upon which to base what he has called his "Experimental Theory of the Steam-Engine;" meaning by this a philosophy established upon a basis of experiment as distinguished from the, in one sense, rational but restricted theory of the ideal engine of pure thermo-dynamics.† Messrs. Isherwood and Emery, and others on this side of the Atlantic, corroborated the work of these first explorers of what was then a new field of scientific investigation; and their explanations of the real physical nature of the com-

* See Rankine's *Miscellaneous Papers; Clausius' Warmetheorie*, 1856; Clark's *Railway Machinery*, 1851–55.

† See Hirn's *Thermo-dynamique*, Paris, 1876.

plex phenomena which characterize the working of every real steam-engine, as distinguished from the thermo-dynamic ideal, have finally given us a perfectly clear and unquestionable knowledge of the subject, and have furnished the basis as well for a correct applied theory.

The enormous discrepancies between the computed efficiencies of the steam-engine, as obtained through the processes devised by Rankine and Clausius, and the actual efficiencies attained by the real engine, even under the most favorable circumstances—differences amounting often, as stated by Hirn, to 30 or 50, or even at times to 60 or 70 per cent.—were immediately accounted for when these experimenters had traced the progress of heat-energy through the machine, and had shown the method of its disposition not only by transformation into work and by necessary thermo-dynamic waste, but also by physical wastes and by transfer by absorption and storage with later rejection and loss, without transformation into mechanical energy. The explanations of Clark, of Hirn, and of Isherwood—who, in his *Researches in Engineering,** gives a very complete and beautiful description of the latter operation, as illustrated by his own experiments—although they were disputed very earnestly at times, and by some of the ablest men of science in Europe and the most experienced engineers in both countries, became accepted immediately upon the conclusion of a discussion in which Hirn and Hallauer took the leading part on behalf of the modern view; and pure thermo-dynamics became an admitted essential element in the theory of the heat-engines, while the science of thermal physics was also permitted to assume its rightful place as being no less essential as one of these two great divisions of the philosophy of the steam-engine.

The results of twenty years of patient and skilful work in experimental research on the part of Hirn, assisted by his able lieutenants, were summarized in his great work on thermo-dynamics, published in 1876, and it was at about this time that the world began to perceive their scientific value and their importance in application. During the next two years we find traces of the application of the clean-cut methods of Hallauer in mathematical discussions, and those distinguished allies of the great physicist and engineer, Messrs. Leloutre, Grosseteste, and Dwelshauvers-Dery, appear, reënforcing his party, in 1875. The last-

* Philadelphia, 1860.

named engineer and scientist began his acquaintance with M. Hirn by friendly criticism of the work of the latter as early as 1873, and joined the committee of self-constituted aids in 1875, for the purpose of repeating and checking the work of the revealer of the then novel facts relating to heat-conversion in the heat-engine. Hirn's statements and views were fully confirmed, and the world has since accepted them. In promoting this acceptance, the work of M. Dwelshauvers and his comrades had no small effect.

Hirn had thus begun this work at an early date and in a purely scientific spirit. He had noted the discoveries of that great genius, Mayer, in 1842, and of his more renowned successor, Joule, only after he had himself endeavored to effect a solution of the very same question: Is heat a substance, or is it a form of energy capable of and requiring transformation in the production of its observed mechanical effects? He had reached the same and the correct conclusion, and in 1848 published his deductions, of which he only became aware of the previous proof several years later, in 1854. He had proved them true in the case of friction; they, for the general case. Immediately thereafter he took up his now famous experiment, proving the conversion of heat into work in the steam-engine, in its regular operation, and on a large scale.

Hirn published his final results in 1855 and later,* and was able to show not only that the ideas as to the nature of heat assumed by Carnot in his then comparatively unknown treatise were incorrect, but also that his own idea, that now accepted, was right, and that heat disappears, as such, in the course of the passage of heat-energy through the engine. He also proved the value of the steam-jacket on his own engine, which, as he stated, furnished 108 H. P. with it, and but 82 without it. On the former point he says that his experiments "completely confirm the modern theory. . . . Heat in a steam motor is not only dispersed, but also disappears; and the power obtained is exactly proportional to the quantity of fluid which disappears as heat to reappear as motive power." Hirn considered Mayer's ideas completely sustained and verified. He then went on with his work in this direction, proving the accuracy of the mathematical prediction of Rankine and of Clausius, that steam must condense, ordinarily, while doing work by expansion. and

* *Bulletin de la Société Industrielle de Mulhouse.*

showed, also, that some vapors, as sulphuric ether, must superheat under similar circumstances. It was this series of experiments, also, which led Hirn to become the earnest advocate of the use of superheated steam. In his hands it proved a grand success.

During this experimental period of Hirn's labors, he succeeded in constructing a theory of the action of the engine which was satisfactory to himself, as to other practical engineers and to those interested in, and familiar with, the applied sciences. This was published in his great work on thermo dynamics, in 1876. In it are presented, for the first time, the theory and the science of the real steam-engine, as distinguished from the ideal engine treated of by the writers on pure thermo-dynamics. It remains to-day the only treatise of that character, although portions of the field have now been traversed by Cotterill in England, and by Ledieu and others on the continent of Europe. But this expression of the Hirn theory and system was still somewhat abstruse to the average engineer, as to the average mathematical reader, and it remained for some one familiar alike with the ideal and the actual case to formulate and sum up Hirn's theory as a general science of the heat-engine.

This was done, in 1878 and later, by M. Dwelshauvers-Dery, a distinguished professor in the School of Mines of Liège, Belgium, now an honorary member of the American Society of Mechanical Engineers. These articles appeared in the *Revue Universelle des Mines de Liège* (1878–80), beginning with an historical sketch of the labors of Hirn and his colleagues in the experimental and theoretical work of developing a "Practical Theory" of heat-engines, and then followed an "*Exposé succinct de la Théorie pratique des Moteurs à Vapeur*" (1880), in which, as admitted by Zeuner, then its most formidable contestant, was for the first time given a thoroughly scientific general theory of the real engine. This work, performed hand in hand by the two philosophers, welded an acquaintance into firmest friendship, and their constant fellowship in spirit and in their common tasks continued to the end of Hirn's long and fruitful life.

The mathematical and algebraic work of reduction of Hirn's theory of the real working steam-engine was thus finally the task of M. Dwelshauvers. This work began, on the part of the latter, as early as 1873, when he first had the opportunity to

make the acquaintance of his great preceptor. Hirn's work was then but little known. His efforts toward the experimental development of the science of the heat-engines date, properly, as far back as 1855; for in that year he presented his first memoir on steam-jackets to the Société Industrielle de Mulhouse. Messrs. Leloutre and Hallauer repeated Hirn's experiments; but no elaborate and satisfactory series of trials of the engine, or any similar, were made until, in 1873, the commission above named was formed, out of pure scientific interest in the matter, and M. Dwelshauvers and his colleagues made a very careful and thorough confirmation of the earlier work. From 1873 to 1878 Hallauer worked with and for his chief, with most admirable and fruitful effect. Dwelshauvers' history of this work, and of the subject to the date of his paper, gives a carefully prepared account of the gradual recognition of the effect of the metallic walls of the working cylinder in the waste of heat and steam, and of the gradual recognition, also, of the value of Hirn's labors, as well as those of Clark and of Isherwood and later writers. The paper was not, however, generally known, having been left in its original form in the transactions of the society mentioned, and never put into permanent shape by publication in book form.

The next work, in this direction, by Dwelshauvers, was the preparation of his "formulated theory," as given in his *Exposé* (1880).* It was this work, perhaps quite as much as the personal statements and writings of Hirn, which led to the now famous discussion, in which Professor Zeuner, himself a great authority in thermo-dynamics, led an attack upon the "experimental theory" by asserting that it must be the mist suspended in the steam, and the water condensed in the cylinder of the steam-engine, which so seriously invalidated the thermo-dynamic theory as applied to the real case. Thus opened the campaign between the advocates of the respective merits of "iron and water as wasters of energy." This discussion led to the still further development of his algebraic theory by Dwelshauvers, who thus brought it into its existing form (1882)—a form which is, as far as it goes, satisfactory to the practitioner as well as to the savant. It had hitherto been the fact that the so-called theory of the steam-engine had been as purely an abstraction, and as absolutely useless to the engineer, as is the pure theory

* *Revue Universelle des Mines, Liège,* 1882, *et seq.*

of the motion of a particle. It required to be supplemented by the physical theory of the various methods of heat-transfer and heat-waste, occurring in the real engine, to give it any value for practical purposes, or to make it satisfactory as well as interesting to the practitioner. These elements were supplied by the experiments of Clark, Hirn, and Isherwood, supplemented by the approximate mathematical treatment of Hirn and Cotterill, refined by the later work of Dwelshauvers.

The latter introduced new matter, as well as reconstructed the older work of Hirn. The great investigator had studied the action of the cylinder and of the engine, stroke by stroke, forward and backward, in the production of internal wastes. Dwelshauvers divided the cycle into four phases, studying each in turn, and thus analyzing more completely the methods of flow and of heat-transfer, producing that now familiar discrepancy between the predicted results of thermo-dynamics and the actual performance of the real engine. Admission, expansion, exhaust or eduction, and compression, were thus made the subject of separate investigation, and equations were produced, exhibiting, each by itself, the quantities of heat-movement as effected through the steam and through the conductivity of the metallic surfaces as well. This was a new and important step, and, so far as is now known, was entirely the work of Dwelshauvers. This gave the means of exhibiting the net action of the waste-producing elements throughout a cycle of the engine. The heat-transformations effected by the action of the steam as working fluid had been noted by Hirn in 1855; but of the six equations now presented in the works of Dwelshauvers, five were of his own and original construction. The "experimental theory of the steam-engine," as it was denominated by Hirn, was apparently first conceived by that distinguished engineer, and the theory was by him fully explained; while the algebraic, the formulated theory, as it to-day stands, is due to Dwelshauvers-Dery.

A graphical analysis of the processes thus formulated was devised by Dwelshauvers-Dery in or before 1888, and was published by him in that year, and in the hands of Professor Unwin, of Madamet, of Sinigaglia, in England, in France, and in Italy, it has been already made to exhibit with greater clearness than ever before what are the conditions which obtain in the working cylinder of every heat-engine, and which distinguish the real from the ideal case.

Dwelshauvers has done much, also, to popularize, in Europe, the modern and accepted theory of the real engine. He has endeavored to exhibit the action of the steam jacket, to show what is the modification of the action of the metallic interior of the engine, by the introduction of that wasteful element to counteract in many cases a greater waste, and he has sought to show the influence of the experimental philosophy of the engine upon the proportions and the working of the condenser. He has observed the fact of a maximum ratio of expansion appropriate to the condition of maximum efficiency, as determined by the variation of this waste, previously unobserved, and is engaged in the construction of its theory in accordance with his published theory of heat-expenditure, reducing all to a common basis and philosophy. In common with all experienced engineers, he hopes to see, ere long, a working theory of the heat-engines so complete that it may serve for a working theory for the designer and the constructor, as well as for the user of the engine.

Some of this work of Dwelshauvers-Dery has been translated by Mr. Donkin, and published, from time to time, in *London Engineering;* other portions remain untranslated, and are only to be found in the *Revue Universelle des Mines;* but its essential portions, it is hoped, may be hereafter presented to the English-speaking members of the profession through his papers for the American Society of Mechanical Engineers.

It is thus that, mainly, or at least very largely, in the works of Hirn from 1855 to date nearly, and in those of M. Dwelshauvers from 1880, that we must to-day look for the formulated theory of the steam-engine, of any real heat-engine.

M. Dwelshauvers presents his algebraic treatment of the operations studied and described by Hirn in the following form: *

I. In any steam-engine, the fluid arrives at the cylinder charged with a certain number of calories of heat, of which it surrenders a portion, then passes on into the condenser, there heating the water, supplied at a lower temperature. It brings in Q calories and retains c calories, which latter are finally distributed to the water of condensation. The difference, $Q - c$ calories lost *en route*, are composed of: (1) T units which serve to produce, by transformation, the external work of the engine;

* *Revue Universelle des Mines, Liège*, 1880.

(2) E units lost by radiation and conduction externally; (3) C units distributed in the condensing water, supplied, originally, at a lower temperature.

If there is a "jacket," the condensation taking place within it gives to the steam in the cylinder another quantity, q units, of heat, which must be added to Q, that supplied by way of the steam-pipe.

Thus we obtain, obviously, the following equation, simply assuming that no heat is either created or annihilated in this process:

$$Q + q - c = T + E + C;$$

and

$$Q + q = T + E + C + c \quad . \quad . \quad . \quad . \quad . \quad (1.)$$

Or: the sum of the quantities of heat imported into the cylinder is equal to the sum of those equivalent to the external work, the heat expended upon the surrounding air, that given up to the condensing water, and that remaining, finally, in the water of condensation.

This is a fundamental principle deduced from our knowledge of thermo-dynamics.

II. But if the steam coming from the boiler is subject to change of temperature and of physical state, some other part of the system must also be subject to complementary variations of temperature. This variation is what is observed in the surrounding walls of the steam-cylinder. But these surfaces must always return, at each completion of a cycle of kinematic changes, to their initial state, and precisely; for, otherwise, the cylinder would either rise or fall in temperature, as the engine continues its operation, and indefinitely. We may hence conclude, according to Hirn and Dwelshauvers:

"The total quantity of heat received by the metallic wall of the cylinder is equal to the sum of those quantities restored by it in the same cycle."

Translating this principle into symbols, we have the following:

The action of the system, at each stroke, includes:

E units of heat surrendered to surrounding objects by conduction and radiation.

q units received by the walls of the cylinder by condensation of steam within the jacket.

R_a units yielded to the walls by the steam during its admission, on coming in contact with the comparatively cold metal.

R_e units restored by those walls during expansion, and taken up again by the expanding steam.

R_c units surrendered back by the walls of the cylinder during the period of condensation, thus sending forward into the condenser substantially all, ordinarily at least, of the fluid, as steam, into the condenser.

Then we have, as the action of the walls:

$$q - E + R_a - R_e - R_c = O;$$

or,

$$q + R_a = E + R_e + R_c. \quad . \quad . \quad . \quad . \quad . \quad (2.)$$

But the three quantities R are related through the physical conditions which follow:

A certain quantity of heat, Q, is brought from the boiler. At the close of the admission period this is partly condensed, and a smaller quantity, U_0, remains. At the close of the expansion period there remains but U_1 units of heat; and, when the exhaust is concluded, we have, finally, in the clearance spaces, a small proportion of the steam, in which is stored but U_2 units. The total quantity of heat which has taken part in the operation of a cycle is thus $Q + U_2$. Now let

T_a units measure the heat equivalent to external work during admission.

T_e units, the heat equivalent to external work of expansion.

T_c units, that expended during compression, and having, of course, the opposite sign.

Then

$$T = T_a + T_e - T_c.$$

The following principles logically result:

III. The heat U_0 possessed by the fluid at the end of admission is equal to the heat Q brought from the boiler, plus the heat U_2 of the steam already filling the clearance and "dead" spaces, minus that which corresponds to the external work of admission, T_a, and minus R_a, that which is yielded to the cylinder walls during the same period.

Or,

$$U_0 = Q + U_2 - T_a - R_a. \quad . \quad . \quad . \quad . \quad (3.)$$

IV. The heat remaining in the fluid at the end of expansion, U_1, is equal to the heat, U_2, at the beginning of that period, plus the quantity, R_e, received from the cylinder walls during expansion, and minus that, T_e, equivalent to the external work of expansion.

Or,

$$U_1 = U_0 + R_e - T_e. \quad . \quad . \quad . \quad . \quad . \quad . \quad (4.)$$

V. The steam at the beginning of the exhaust period contains U_1 units of heat. Now, R_c is received from the cylinder walls during the period of exhaust; T_c comes from the external work of compression; but C units are surrendered to heat the condensing water, and c only is left after its own condensation; while the remainder, U_2, is that left in the clearance spaces for the next stroke. Whence,

$$U_2 = U_1 + R_c + T_c - C - c. \quad . \quad . \quad . \quad (5.)$$

Only three of the quantities which enter these five equations are necessarily obtained by experiment; these are R_a, R_e, R_c, which relate to the internal transfers of heat, produced by the action of the cylinder walls.

Three of the five equations may therefore be made to determine the three unknown quantities, and the other two will check the experiments.

It will thus evidently be convenient to rearrange the equations thus:

$$\begin{aligned}
R_a &= Q + U_2 - U_0 - T_a. \quad . \quad . \quad . \quad . \quad (A.)\\
R_e &= U_1 - U_0 + T_e. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (B.)\\
R_c &= U_2 - U_1 - T_c + C + c. \quad . \quad . \quad . \quad (C.)\\
R_a - R_e - R_c &= E - q. \quad . \quad . \quad . \quad . \quad . \quad (D.)\\
Q + q &= T + E + C + c. \quad . \quad . \quad . \quad . \quad . \quad (E.)
\end{aligned}$$

These equations are perfectly general, and apply to any kind of heat-engine and to steam-engines, simple or compound, with or without jackets, and with or without superheating.

It is readily seen that equation D shows that, if the external radiation may be neglected, the sum of the quantities of heat supplied by the jacket, q, and by the steam, R_a, is equal to the sum of the quantities restored by the surfaces of the cylinder, to

the steam, and usefully, during the expansion, R_e, and that lost entirely during condensation, R_c. It is, therefore, the relation of R_c to $Q + q - c$ that determines the relative loss due to the action of the walls, in engines subject to such comparison, and which thus furnishes a means of comparison of their relative economical standing.

This brief analysis and exposition of the work of our colleagues, the great engineer-physicist who has just left us and his able disciple, will serve to show the general methods of treatment of this only recently understood process of internal waste; and the illustration of their practical application may be sought in the papers of Hirn, in his treatise on thermodynamics, in the contributions of Hallauer and of Dwelshauvers to the *Bulletin de la Société Industrielle de Mulhouse* and the *Revue Universelle des Mines*, and, we may perhaps venture to hope, in those to be presented by the latter to the American Society of Mechanical Engineers.

DISCUSSION.

Prof. H. W. Spangler.—In the paper by the author, read at the New York meeting, so much of a topic cognate to that herein treated was discussed by the Society, that I should not care to reiterate the opinions, expressed at that time, for publication in the same volume of transactions.

The historical part of the paper is of interest, but the mathematical treatment of the subject of the real engine, follows very closely the lines of the previous discussion.

*Prof. R. H. Thurston.**—This subject is so important, both historically and from a scientific point of view, that I feel sure that I may be excused if I have taken a little space for it. It is recognized by Zeuner, and all later authorities as the most important of recent extensions of the philosophy of the heat-engines.

I find that the mistake has sometimes been made of confounding the work of Dwelshauvers-Dery with that of Zeuner himself, instead of recognizing the latter as indebted to the former for its first expression, as has been shown, in detail, by Professor Schmidt, of Prague; the notation, however, having been altered to correspond with that customarily adopted by that writer. (See *Revue Universelle des Mines*, t. XII., 1882.)

*Author's closure, under the Rules.

CCLXXXIII.

DETERMINATION OF THE SENSITIVENESS OF AUTOMATIC SPRINKLERS.

BY A. F. NAGLE, CHICAGO, ILL.
(Member of the Society.)

AUTOMATIC sprinklers for the extinguishment of fires are introduced to such an extent now that a scientific analysis of their sensitiveness is a necessity in order to form a correct judgment of their true action. Let it be premised that all modern automatic sprinklers use the same low-fusing solder (155° Fahr.) for holding their outlets closed. Yet the time and temperature at which the solder fuses, which secures the valve in place, vary considerably with different makes of sprinklers. This difference in time and temperature is due to the quantity of solder to be melted, the amount of metal with which the solder is in actual contact, the proximity of the solder joint to large masses of metal, and the strain upon the solder.

The purpose of this paper is to correct the erroneous opinion now held as to the *actual* temperatures at which automatic sprinklers open.

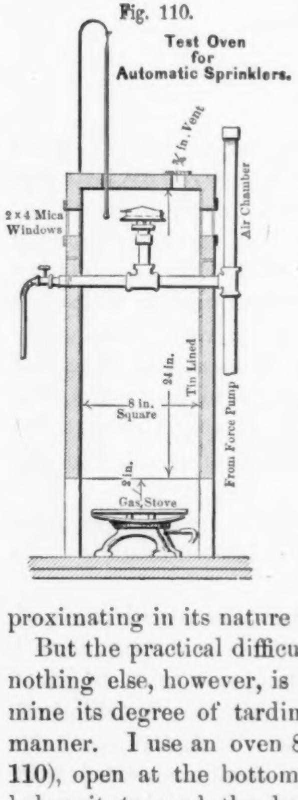
The method generally pursued to express the sensitiveness of sprinklers is to place the sprinkler to be tested in an oven, heated by a common gas stove, and note the temperature at which it opens. Such tests, so broadly stated, give rise to very different published statements of their sensitiveness. There are many details connected with a scientific test which should be stated with precision in order to make tests made by different parties comparable. Ovens are made of wood, lined with tin, iron, and glass; and the sizes of ovens vary from 1 cubic foot to 2,400 cubic feet; and the time during which the test continues is never mentioned. The reading of the mercurial thermometer to a fraction of a degree, however, is regarded as all-important, and taken to be a conclusive and correct indication of the sensitiveness of the sprinkler. The writer having observed these published results for many

years, and made many experiments himself, has become convinced that the mercurial thermometer is a very sluggish means for correctly indicating such rapid-changing temperatures as are going on within an oven heated by burning gases.

Air carries such a small volume of heat that it must be heated to quite an excess of temperature above that indicated by the thermometer, in order to make the thermometer ascend at a rate corresponding to the conditions of a fire. That air is such a delicate carrier of heat may be best understood when we consider that it contains less than $\frac{1}{3000}$ as much heat for the same volume as water. To demonstrate this, a cubic foot of water at 62° Fahr. weighs 62.3 lbs., and a cubic foot of air at the same temperature 0.0761 lb., a ratio of 819 to 1; and as the specific heat of air at this temperature is only 0.2377, the relative volumes of heat contained in the two substances are as 3,025 to 1.

Fig. 110.

Test Oven for Automatic Sprinklers.



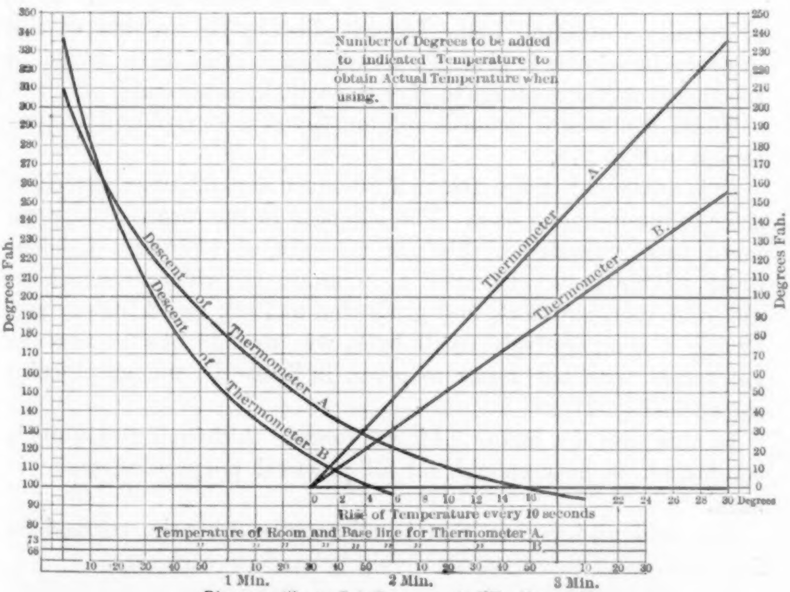
A hot-water test would not be delicate enough, for no appreciable difference in the sensitiveness of sprinklers would be discernible by a water test. Air remains as the only practicable agent, and the one most nearly approximating in its nature to actual fires.

But the practical difficulty is in the mercurial thermometer. As nothing else, however, is feasible, the only thing to do is to determine its degree of tardiness. This I have done in the following manner. I use an oven 8 inches square and 24 inches deep (Fig. 110), open at the bottom, and a small gas stove placed 2 inches below it, to send the hot gases upward. The cover joint is $4\frac{3}{4}$ inches from the top, and the thermometer projects 2 inches into the top. Two small vent-holes ($\frac{3}{4}$ inch) in the top can be opened or closed to permit of different rates of heating the oven. The sprinkler is screwed into a cross-pipe, leaving the solder joint about 2 inches from the top, or on a level with the thermometer bulb. The heat from the gas stove opens different sprinklers in from 1 to 3 minutes, and at *indicated* temperatures ranging from

190° to 300°. This time may be considered the same as usually operates in actual fires. Having observed the rise of temperature in the oven every 10 seconds, its varying rate is obtained and recorded in Table A and graphically illustrated in Fig. 111 for a slow-oven test, and in Table B and Fig. 112 for a quicker oven test. Column 2 in each case gives the rate, or number of degrees; the temperature increases every 10 seconds.

To determine the corrections, the thermometer was heated over

FIG. 113. FIG. 114.



Diagrams illustrating the amount of Tardiness of Two Mercurial Thermometers under different rates of rise of Temperature.

the gas stove to 300° or so, and instantly removed, and its fall of temperature noted every 10 seconds in the room, which was at 73° Fahr. Some experience was necessary to accomplish this satisfactorily. If the thermometer were allowed to hang still, the air would be heated above the normal temperature of the room (73°), and hence no correct knowledge of the surrounding temperature about the thermometer would be had. It became necessary, therefore, to fan the air quite violently in order to maintain the temperature about the thermometer at 73°. At first thought this may seem like cooling the thermometer, but that is not so—no fanning

can reduce the air below the temperature of the room. The circulation of the air about the thermometer in the room may also be analogous to the circulation of the hot gases in the oven, produced by the natural ascent due to their great heat. Table C and Fig. 113 give the readings of the thermometer, and column 2 its rate of decrease. Column 3 is the difference between the temperature of the thermometer and that of the room. Column 4 is the mean difference every 10 seconds, and column 5 is the mean difference divided by the amount of fall every 10 seconds.

The factor obtained in column 5 is an important one. It is practically a constant, 7.80. If multiplied by the amount of rise in every 10 seconds it gives the number of degrees to be added to the thermometer readings to give the actual temperature.

These results are given in column 3, and are graphically represented in Fig. 114.

It is evident that the factor 7.80 must depend upon the character of thermometer used. The one used in the foregoing experiment was graduated to 350°, on a length of 14 inches, and had quite a large bulb. In order to ascertain the different effect on different thermometers, one graduated to 700°, on 14 inches, and having a comparatively small bulb, was used in like manner as heretofore, with the result as given in Table D, and graphically shown in Fig. 113.

In this case the factor 5.44 was found. Although not quite a constant, it is near enough for all practical purposes to make it such.

It is evident, however, that in all accurate tests the corrections for tardiness of thermometer should be made in each case.

Fig. 114 is a graphical representation of the amount to be added to the indicated temperature for every probable rate of increase of rise, in order to obtain the actual temperature.

It is certain that temperature alone is not a correct expression of the actual sensitiveness of automatic sprinklers, even if correction be made for the slowness of the mercurial thermometer in responding to the actual increase of the temperature of the hot gases. The area inclosed between the actual temperature line, the time line, and the temperature of the room line is, probably, as nearly correct an expression as is possible to obtain. This area might be termed *Thermal Minutes*, and in this manner each sprinkler could obtain a factor for thermal minutes which would convey some intelligent idea of its true sensitiveness.

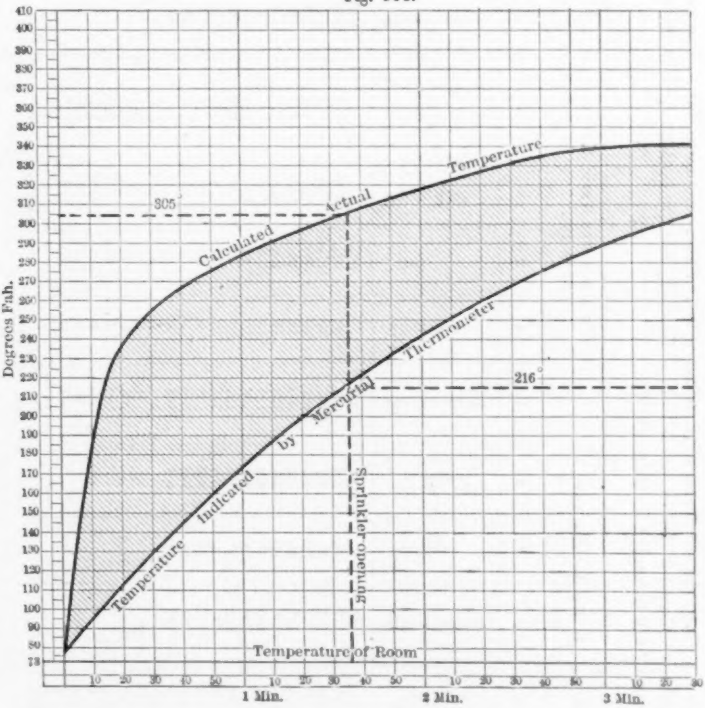
EXPERIMENTS TO DETERMINE THE *ACTUAL* TEMPERATURES IN AN OVEN FROM THE APPARENT ONES INDICATED BY A MERCURIAL THERMOMETER.

The thermometer used was a high-grade thermometer, graduated to 1° Fahr. and up to 350°. The oven was 8″ × 8″ × 24″ deep, with the upper 4¾″ removable, a gas stove placed 2″ below the bottom of the box, and the bottom open to the air. The oven was made of wood and lined with tin. Thermometer bulb extended 2″ through top.

TIME.	A. SLOW-OVEN TEST.			B. RAPID-OVEN TEST.			C. THERMOMETER TEST, A. ROOM 73°.					D. THERMOMETER TEST, B. ROOM 68°.				
	1	2	3	1	2	3	1	2	3	4	5	1	2	3	4	5
Minutes and Seconds.	Thermometer.	Increase.	Actual Temperature.	Thermometer.	Increase.	Actual Temperature.	Thermometer.	Decrease.	Diff.	Mean Diff.	Col. 4. / Col. 2.	Thermometer.	Decrease.	Diff.	Mean Diff.	Col. 4. / Col. 2.
0......	79	2	....	95	....	....	307	....	234	....	....	336	....	268	....	....
10......	91	12	185	111	16	236	277	30	204	219	7.3	284	52	216	242	4.65
20......	110	19	258	131	20	287	247	30	174	189	6.3	243	41	175	195	4.75
30......	128	18	268	151	20	307	227	20	154	164	8.2	212	31	144	159	5.10
40......	144	16	261	171	20	327	209	18	136	145	8.	186	26	118	131	5.04
50......	159	15	268	191	20	347	193	16	120	128	8.	165	21	97	107	5.10
1.00......	173	14	282	210	19	358	179	14	106	113	8	148	17	80	88	5.18
10......	186	13	287	228	18	368	166	13	93	100	7.6	135	13	67	73	5.61
20......	198	12	292	245	17	378	154	12	81	87	7.2	124	11	56	62	5.64
30......	210	12	304	261	16	386	144	10	71	76	7.6	115	9	47	51	5.64
40......	221	11	307	276	15	393	135	9	62	66	7.4	108	7	40	43	6.14
50......	231	10	309	290	14	399	128	7	55	59	8.3	102	6	34	37	6.17
2.00......	241	10	319	303	13	404	122	6	49	52	8.7	96	5	28	31	6.32
10......	250	9	320				116	6	43	46	7.7					
20......	259	9	329				111	5	38	41	8.1				Avg...	5.44
30......	267	8	329	Avg. Inc ...	17.3		106	5	33	36	7.1					
40......	275	8	337				102	4	29	31	7.7					
50......	282	7	337				99	3	26	28	9.2					
3.00......	289	7	344				96	3	23	25	8.3					
10......	295	6	342				93	3	20	22	7.3					
20......	300	5	339													
30......	304	4	335				Avg. Dec...	11.3		Avg...	7.8					
40......																
50......	Avg. Inc....	10.7														

To illustrate the frequent misuse of these temperature tests, let us take the case of four sprinklers of two different patterns. A party subjects his own sprinkler to an oven test where the rise of temperature is such as to open the sprinkler in 1 minute 35 seconds (see Fig. 113); however, he takes no notice of this time, and reads the thermometer at the precise instant the sprinkler opens, at 216°; but the *actual* temperature is about 305°. He then places the sprinkler of the other pattern in the same oven, but owing to the fact that, even

Fig. 111.

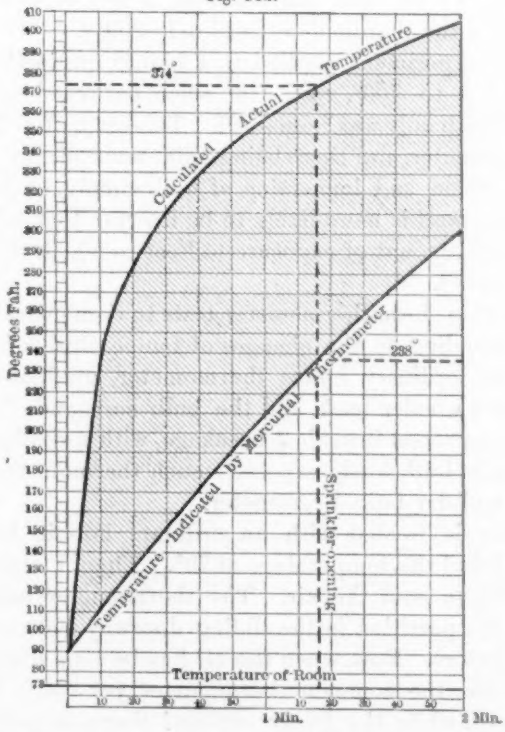


SENSITIVENESS OF SPRINKLER—290 THERMAL MINUTES.

with considerable well-intentioned care, the oven and the thermometer have not cooled to the same temperature as when he tested his own, the oven now heats up quicker (see Diagram II.), and while the sprinkler opens in 1 minute 16 seconds (again he does not note the time) the temperature is noted with exactness at 238°, but actually about 374°.

Now, this extra *apparent* 22° attributed to this latter sprinkler is a very important and damaging amount with which he goes before the public in selling sprinklers. On the other hand, another party may make just such a test for himself, giving his sprinkler, however, the advantage of a slower test, and find just the same result, but in this case in favor of *his* device. Now, if the actual thermal minutes had been computed in each case, no appreciable difference would

Fig. 112.



SENSITIVENESS OF SPRINKLER—290 THERMAL MINUTES.

have been found in these two different sprinklers. One would have given 292 thermal minutes and the other 290. This illustrative case is neither a fanciful nor an extreme one, but such as happens to well-meaning parties. It shows what injury and injustice may be done to innocent parties by reputed scientific tests of sprinklers by underwriters, inspectors, and sprinkler manufacturers, ignorant of thermal laws.

In inserting a thermometer into a test oven the bulb only is acted upon by the heat. I do not know whether thermometers are graduated by entire immersion, or by immersion of the bulb only.

In order to ascertain the exact facts, I immersed a thermometer (A), graduated to 350° Fahr., in boiling water up to the 200° point and found it read 212°. Then the bulb only was immersed and the part above was kept at the temperature of the room by surrounding the exposed portion with a wet cloth of 70° temperature, then the thermometer read only 209½°.

A similar experiment applied to a higher grade thermometer (B), graduated to 700°, read 216° when entirely immersed, and 215½° when bulb only was immersed. This error of 4° in the high-grade thermometer, and there being a difference of only ½° between entire immersion and immersion of bulb simply, shows that the thermometer itself is more likely to be in error than what may be caused by the amount of exposure to heat to which the thermometer is subjected.

Since making these thermometer tests to demonstrate its slowness in responding to the changes of temperature of air, I have compared an ordinary house thermometer, graduated to 130°, mounted on a wooden back, and the bulb surrounded by a light brass cage to protect it against breakage, with a similar thermometer having its bulb freely exposed—such thermometer as is used in the wet and dry bulb hygrometer.

My house is heated with an ordinary hot-air furnace, and frequently I find the temperature at 79°. Then I open the doors and windows to cool the air. The thermometer hangs against the end of the partition in the sliding doorway between the front and back parlors. Now, when the air has become so cool as to be uncomfortable, the common thermometer yet reads 74°, having fallen only 5°, while the freely exposed thermometer reads 68°. Since discovering this I have heard friends mention the awful slowness of their metallic thermometer under similar conditions.

These facts call our attention to the necessity of correcting the apparent temperatures where the hot air and gases are in a changing state, as it is simply impossible to obtain an instrument which will instantly respond to the changing temperatures of hot gases.

It will be seen from the foregoing that a thermometer does not give the correct temperature at which a sprinkler opens, and

that the exact temperatures can only be obtained by calculations based upon experiments as indicated in this paper.

The *relative* sensitiveness of sprinklers, however, can be expressed without resorting to this correction, provided that the same apparatus be used for each and all tests and that the apparatus be used with the same care.

For the purpose of showing what accurate results can be obtained with the simple oven illustrated in the paper, I will state the mode of procedure and results obtained in experimenting with two sprinklers of different make, say A and B. In order to reduce the chances of errors of conditions to a minimum, the sprinklers were tested alternately. No pressure was put upon the sprinklers, in order to avoid wetting the oven, as the spraying of water into it would be likely to disturb the uniformity of the oven temperatures. This would affect neither of the sprinklers used, and, in fact, nearly all sprinklers have their levers under sufficient strain, in keeping the valves tightly shut, that they act as a spring to sever the solder joint. Hence, a sprinkler can be tested without the use of a force pump, and thus remove one cause productive of inaccurate oven conditions. The pop with which sprinklers release will be noticed when the sprinkler is tested dry, just the same as when it is tested under pressure.

The thermometer used was graduated to 700° Fahr., called thermometer B, in Fig. 113, but probably $3\frac{1}{2}$° in excess at the boiling point as shown.

In order that the gas flame might be uniform, it was not extinguished after it was once lighted, but the oven was bodily removed at the conclusion of each test, and restored at the beginning of the next.

There was a pause of five minutes between each test, so that the oven could cool alike each time. The thermometer was cooled in water to 66°, and then inserted in the oven until it remained nearly stationary somewhere between 80° and 86°, when the test began.

Three preliminary tests of the oven alone were made so that uniformity of heating might be established.

SPRINKLER A.

No. 1.	Temp.	Time.
	80°	0″
	100	10
	140	26
	180	44
	200 Opened.	55

No. 3.	Temp.	Time.
	86°	0″
	100	9
	140	25
	180	44
	200	55
	220 Opened.	1′ 6

No. 5.	Temp.	Time.
	85°	0″
	100	9
	140	24
	180	43
	200	51
	206 Opened.	54

No. 7.	Temp.	Time.
	86°	0″
	100	8
	140	23
	180	40
	200	49
	204 Opened.	52

SPRINKLER B.

No. 2.	Temp.	Time.
	84°	0″
	100	10
	140	25
	180	43
	200	55
	220 Opened.	1′ 5

No. 4.	Temp.	Time.
	83°	0″
	100	10
	140	27
	180	46
	200	57
	212 Opened.	1′ 4

No. 6.	Temp.	Time.
	86°	0″
	100	10
	140	27
	180	45
	200	55
	218 Opened.	1′ 6

NOTE.—Test of thermometer in boiling water indicates that it registers $3\frac{1}{2}°$ too much at the boiling point.

AVERAGE.

Temp.	Time.	Temp.	Time.
$207\frac{1}{2}°$	57″	217°	65″

It will be observed that with the care exercised, exceedingly uniform oven action was obtained, and hence, the tests are considered a fair and correct comparative measure of the two sprinklers for sensitiveness.

NOTE.—Correcting for error of thermometer at boiling point, $3\frac{1}{2}°$, Sprinkler A opened at 204° and Sprinkler B at $213\frac{1}{2}°$ apparent temperature. These figures could be used in comparison with other sprinklers if tested with the same apparatus in the same careful manner. In order, however, to illustrate the method of obtaining the corrected temperature, an examination of the recorded temperatures shows that the temperature was rising at the time of opening of the sprinkler, at the rate of about 20° in 10 seconds. We had found the constant for the thermometer used (B) to be 5.44, hence the correction to be applied is 20 x 5.44 = 108.80°, or say 312.80 for Sprinkler A and 322.30 for Sprinkler B. In order to obtain the thermal minutes required for each sprinkler, the reading of the thermometer should have been recorded every 10 seconds, and a diagram constructed as shown in Fig. 112, the area of which figure would be the nearest expression of the actual sensitiveness of the two sprinklers. Pursuing that method illustrated in Fig. 112, as well as possible from the data given, we find that Sprinkler A re-

quires 170 and Sprinkler B 205 thermal minutes, an excess of 26%. This is a more comprehensive method of expressing the sensitiveness of automatic sprinklers than has hitherto been the practice.

DISCUSSION.

Prof. D. S. Jacobus.—A number of tests of automatic sprinkler heads have been made by me, under the direction of President Morton, of Stevens Institute of Technology, for the New York Board of Fire Underwriters, and other parties.

The sensitiveness of a sprinkler head is only one of the elements which must be considered in determining whether it is perfect enough to be adopted. In fact, a small difference in the sensitiveness of two different forms of heads will be in many cases much more than counterbalanced by some other advantage which the less sensitive head may possess over the other. In addition, therefore, to describing our method of making tests to determine sensitiveness, a brief account will be given of the other tests regularly carried out, in order that one may more readily appreciate how small a factor of the whole the sensitive test is.

A criticism which may be made on the author's paper is that he infers that it has not been usual in tests made in an oven to note the time required to set off the head as well as the temperature. Our practice will not bear out this statement, for in all tests, some of which were made more than five years ago, the time has been recorded, and included in the report.

From the cut given of the testing oven used by the author, it appears that all the sprinkler heads are placed in a vertical position above the pipe to which they are connected. If the sprinkler is intended to be attached on the under side of the pipe, as is the case with the greater portion of those now in use, it appears more logical to test them in this way, and other advantages will be gained by doing this for reasons which will be discussed later. Again, in Mr. Nagle's tests, air appears to be used to produce a pressure on the head; this, although it conforms to the practice in dry-pipe systems, is not, in our opinion, as good a method in testing as to use water, because the presence of water in certain forms of heads retards their action. It is thought best by us to test under the most disadvantageous circumstances which will be present in ordinary practice, and for this reason water, and not air, is employed.

The experiments showing the difference in the actual tempera-

ture of the air and the reading of the mercurial thermometer made by the author of the paper are useful and interesting. This is not taken into account in our experiments, but is partly eliminated by making the tests of a longer duration than those made by the author. The capacity of the gas-burner used to heat the oven used by us is such that the temperature is raised to 200° Fahr. in about 3 minutes, and after this time the radiation of the sides of the oven is so great that the heat increases very slowly, rising to 230° in about 5 minutes from the start. The average sprinkler operates in about $5\frac{1}{2}$ minutes at 235° Fahr., so that, by the time the head opens, the temperature of the oven is increasing very slowly, and the error due to the difference of the actual temperature of the air and the reading of the thermometer is not a great one. On account of the existence of this small error it was proposed to preserve the oven at a constant temperature, say 250° Fahr., introduce the sprinkler head without sensibly altering this temperature, and note the time required to set off the head by means of a stop-watch. It was, however, decided that the test for sensitiveness as now conducted was accurate enough, and the above refinement was not adopted.

The following tests are gone through by us on each set of sprinkler heads:

1st. Tests in the oven to determine sensitiveness and reliability of action.

2d. Tests by applying direct flame, at intervals, to the head to determine reliability of action.

3d. Tests of distribution and rate of flow of water through the heads.

The most prevalent, and at the same time the most dangerous, defect usually found in sprinkler heads is the sticking of the valve to its seat after the releasing device has operated. To test the heads thoroughly in this respect the first and second tests, in addition to being made on the new heads, are repeated on heads which have been connected to piping and subjected to the action of water and of brine for about three months. The most practical tests are those made on heads which have been in use for several years, but as this cannot be done with a new form of head, the plan of filling with water and brine was adopted. The action of the water and brine for three months has caused the valves of many heads to stick so that they would not open under ordinary pressures.

The oven used for the sensitive tests is 2 feet in height and of 5 cubic feet capacity. At two of its sides, pieces of glass are inserted about 1 foot long by 6 inches wide. A thermometer hangs at such a height that its bulb is about midway between the top and bottom of the oven. An arrangement of swing joints allows the pipe to be drawn outside of the oven, so that the sprinkler head may be readily screwed to its place. Before pushing the pipe back to its position in the oven, a water pressure is made to act on the sprinkler, and any air which may have been in the pipe is removed by means of a vent. The sprinkler is allowed to stand about 2 minutes in order to test for leakage, and then is pushed to its position in the oven. The bulb of the thermometer is placed near the solder joint and at the same height as the latter. The thermometer employed has a translucent scale, which is illuminated by a gas flame placed opposite the window at the back of the oven. A system of plates is used at the bottom of the oven to distribute the heat throughout the same. A gas-burner is employed for heating.

A tank is mounted above the oven at such a height that 10 feet head of water is obtained. The sprinklers are usually tested under this small head of water. If the valve sticks to its seat after the releasing device has operated, a valve leading to the tank is closed and the pressure gradually increased to that in the city mains, in order to determine if this will open the sprinklers. If the sprinkler opens, the exact pressure at which it does so is recorded.

The advantage of employing water in the pipes instead of compressed air, and connecting the sprinklers in the same way as is done in practice, arises from the fact that we can test their reliability of action at the same time that we do their sensitiveness. Sometimes sprinkler heads will fail on account of a slight leak starting at the valve, which allows a small amount of water to run over the partly melted solder joint of the releasing device, thus chilling it and preventing the complete opening of the head. This action would not occur if compressed air were used in the pipes, or if the sprinkler were connected in such a way that the water which escaped by leakage would not run over the solder joint.

The tests made by the intermittent application of a direct flame often show defects in the releasing device, which are not apparent in the oven tests, as there is a greater tendency to start the solder

joint slightly, without completely melting the same, so that in some heads the releasing device partly opens and allows water to leak over the solder joint and chill it, as already explained in connection with the oven tests.

Various other defects are sometimes found in the releasing devices. In some cases a loose piece will fall in such a way that it will become caught in the frame of the sprinkler head and prevent the valve falling to its proper position. In others, the valve is not guided so that it always falls to its proper position. Others have been received in which the amount of force made to bear on the solder joint was so small that the cohesion of the melted solder prevented some of the heads from opening under a light water pressure. Others have opened a short distance, and then become caught at some point in such a way that they will not open even under a heavy water pressure.

The tests of distribution and rate of flow are made under a head of 10 feet as well as under high pressure. A barrel is mounted so that the level of water in it is 10 feet above the centre of the connection into which the sprinkler is secured. A large valve is placed directly under this barrel, by means of which the water may be admitted to the sprinkler without producing a throttling action. A 1-inch ordinary steam-pipe screws into a bushing fitted in this valve. At a distance of 10 feet below the surface of the water a horizontal length of $\frac{3}{4}$-inch pipe 10 feet long is connected by means of an elbow. The reason for employing the lengths of pipe given above was to connect the head so that the water furnished it should have to travel through about the same length of pipe of a small diameter as it would in the case of a sprinkler connected to a system in a building. The sprinkler head connected to the testing apparatus is 8 feet above the floor.

The New York Board of Fire Underwriters require that the distribution 8 feet below the sprinkler head shall be over a circle whose diameter is 10 feet, that the diameter at 6 inches above the head shall be 8 feet, and that the rate of flow shall be greater than 1 cubic foot per minute.

To measure the diameter 6 inches above the head a graduated straight-edge is employed. The observer, standing on the same level as this straight-edge, can readily observe the diameter spread over.

In some tests a flat surface has been placed 6 inches above

the head to show how the water will be distributed when the head is used under a ceiling. If the water is thrown upward at too great an angle it will not distribute properly if used under a ceiling. All the heads are tested under 20 lbs. pressure as well as that due to 10 feet head.

The height of water in the barrel is shown by a pointer which magnifies the difference so that ¼ inch in the height moves the pointer about 3 inches. The rate of flow is measured by a meter.

It was questionable at the start whether the friction produced on the water by the pipes leading to the sprinklers would remain constant. If this varied, all the tests made on the rate of flow would not be comparable. One of the first sprinkler heads received, which was in April, 1885, was, therefore, carefully preserved, and before making any of the succeeding tests the rate of flow for this head is redetermined. The diameter of outlet of the sprinkler is 0.555 inch, and it discharges 1.54 cubic feet per minute. Determining the rate of flow for this sprinkler before making trials of others also serves to check the accuracy of the meter. The standardization of the meter at the time of making the first tests was accomplished by weighing the water discharged by the sprinkler.

An additional test which requires much time to secure satisfactory conclusions is one for liability to loss by water damage. Many heads are arranged so that they may be tightened up if leakage occasions. This tightening up is liable to bring an excessive strain on the solder joint, which may yield, not immediately, but perhaps one, two, or three weeks afterward. It appears that the strain produces a slow weakening action. This action has often been observed in the heads which were connected to open index pipes filled with water and brine, and set aside for three months for the first and second tests, as already explained. Many of the so-called instances in which sprinklers have been opened by the water-hammer may have been simply caused by excessive strain on the solder joint.

In addition to complying with the above tests, there are certain other features which must be considered before adopting a sprinkler of a new pattern. For instance, if a collection of dust or dirt on any portion of the head will seriously impede its action, or if there is a liability that corrosion in the releasing device will in any way interfere with its working

Testing of automatic sprinklers is in many cases only one of the elements which have to be considered before adopting a fire-extinguishing system, as various devices have been adopted to prevent freezing of water in the pipes. Some devices accomplish this end by opening the valve, admitting water to the system by the action of air or electricity; these are called dry-pipe systems. A non-freezing liquid is sometimes used in the pipes. I have examined many of these systems for and together with President Morton, and may, at some future time, present a paper giving the methods employed in inspecting and testing the same.

Mr. Charles E. Emery.—The paper of the author shows very clearly the principles to be observed in the construction of automatic sprinklers, and the principal distinguishing features of apparatus and methods necessary to make proper tests of the same. The writer clearly points out how it is possible to test such devices, and to state correctly the actual results obtained in detail, when, on account of the methods employed, the tests do not show the true comparative value of competing devices. This can be done in testing many other devices than automatic sprinklers. For instance, experts know what house-owners and steam-fitters do not, that the efficiency of a radiator depends upon the freedom with which air can circulate over the radiating surfaces, so that engineers can be found who will make a careful test of a client's radiator with a given surface distributed over considerable length, for instance, with only two rows of small pipes, and test in comparison an opponent's wide, short radiator, arranged, say, to put the same surface in pipes four or five rows deep. The long, thin radiator will condense more water every time, and if the engineer in his report makes a correct drawing of the radiators tested, so that on the face of the report he does not appear to deceive the public, but is careful not to mention the principles involved, the client has an opportunity of extolling the merits of his particular radiator in comparison with his opponent's, and to gain a commercial advantage not entirely warranted by the actual relative values of the goods themselves. It is suggested, as a subject for topical discussion, the inquiry as to what extent engineers are warranted in making and publishing tests of this character.

Prof. R. H. Thurston.—The paper before us is a very interesting one to me, especially from the fact that it gives some data bearing upon an entirely different subject of investigation. The importance of the automatic sprinkler has now become too well estab-

lished to permit us to challenge the opening remark of the paper. They are of enormous importance, and have proved their value on a thousand occasions by saving millions of dollars' worth of property by the prevention of what must, in their absence, have been destructive fires. To procure solder for these instruments which shall certainly melt at a specified temperature, and at any moment, whether a week, or a month, or a dozen years after installation, is an essential to their safe employment. That the indications of any thermometer situated in an atmosphere the temperature of which is rapidly rising, are inevitably and invariably inaccurate, requires no proof. It is a matter of common observation, and is obviously unavoidable in consequence of the circumstances described in the paper. But if we can ascertain just what is the action of the thermometer under specified conditions, it is possible, I imagine, that we may find ways of test which shall give as satisfactory results for their purpose as those given by a chronometer for its purpose; the instrument never indicating the right time, but its rate being known and its initial indications being on record.

Figs. 111 and 112 illustrate admirably, at once, the defect of the usual methods of comparison and the proper mode of investigation and rating. But it should be remarked, I think, that the comparison is not simply between the temperatures of room and thermometer. The rise in temperature of the sprinkler itself is, perhaps, more likely to follow the law illustrated by the thermometer, than that of the variation of temperature of the atmosphere about them. So that it might prove that, in certain cases, the melting temperature of the solder should be rather that indicated by the thermometer than that of the air in the room. That is to say, the melting point of the solder should be something less than that which it is desired to set as the limit of temperature of the air in the room in which that sprinkler is to be placed. How much less would depend, it would seem, upon the character of the construction of the sprinkler, quite as much as upon that of the solder itself. The conductivity, the fusibility, the weight of both solder and surrounding metal, and the extent of surfaces exposed to external heat and to contact between metal of sprinkler and solder—all these, and perhaps still other circumstances, must more or less affect the result and help to determine the value of the sprinkler for a specified place and purpose. Mr. Nagle's investigation will point the path to exact research in all these directions.

There is another direction in which we are, many of us at least, interested, and in which this bit of work affords valuable suggestions. One of the most important problems confronting the engineer interested in the applications of the heat-engines is that of the determination of the laws and the data relating to the internal heat-wastes of those most important of all products of the inventive genius of mankind. But we are there confronted by precisely such a difficulty as is here so well illustrated. We, as yet, have never been able precisely to determine the method of variation of temperature of the inner walls of the cylinder, and the method of storage of heat in them, so as to be able to predict, in any given case, just how much heat and steam, and fuel and money, will probably be wasted by the phenomenon now familiarly known as "cylinder-condensation" in the steam-engine, and, in a slightly different form, no less influential in the production of losses in the other heat-engines. Students of heat-transmission, all over the world, like Fourier and his followers, and engineers making their investigations of the actual economical working conditions of the machine, have been vainly seeking the solution of the problems thus brought before them; but all that we can to-day say is that we can, under certain conditions and for certain cases, approximately estimate these wastes and the resultant efficiency of the engine. Could we secure exact measurements of the temperatures of the successive layers of the interior of the cylinder walls, it is very possible that we might determine with some accuracy the laws of such wastes. This has heretofore been attempted, but with no result. I have sought, in all directions open to me, a solution of this problem; but have not succeeded in obtaining what seemed desirable. I am hoping still to see some outcome of a useful, or at least suggestive, nature from researches now in progress, but have no reason to hope that they will fully meet the requirements of the theory of the real engine, as I have called it, to more than a very limited extent. This revelation of the absolute worthlessness of the mercurial thermometer for any such investigations is in itself likely to prove useful; though I presume that no one would to-day attempt to secure measures of varying temperatures by its use, under the conditions to which I now refer. Mr. Dixwell used a better system; but it looks as if only some modification of the electrical thermometer would give us determinations of any value. I hope to be able to give an account of some such researches, after a time; but I cannot as yet

say that I have much expectation of being able to give such accurate data and determinations as are desired and, indeed, required.

Mr. C. J. H. Woodbury.—The high efficiency attained by automatic sprinklers as a means of extinguishing fires on industrial property has very naturally attracted a great deal of attention to the subject, both on the part of the owners of property, and also on the part of persons of a mechanical turn of mind. Although the first automatic sprinklers were invented early in the century, yet the production of practical devices, and their introduction on a large scale, has been limited to the last twelve years.

The efficiency of such devices is shown by a compilation of the fires occurring on property insured by one insurance company from Jan. 1, 1887, to Jan. 1, 1890. These fires are divided into two classes, the first including those where automatic sprinklers formed a portion of the apparatus in service over the origin of the fire, and the second including those which lacked the service of such apparatus. At 290 automatic-sprinkler fires, only 122 of which reached sufficient amount to be followed by claim, the losses amounted to $127,665.44, the average cost of each fire being $433.32. The other losses amounted to $7,388,503.25 at 947 fires, 400 of which were followed by a claim, the average cost of each fire being $7,802.01, or 18 times the average loss occurring at fires under automatic sprinklers. It is but fair to say, however, that these figures deal with full automatic-sprinkler systems, and from them have been eliminated such instances as where the water was shut off from the automatic sprinklers, or where they were not provided with a proper static supply in the usual manner. An automatic-sprinkler system is not different from other mechanical devices in requiring frequent inspection.

But these results were reached by a very few varieties of automatic sprinklers, although the whole number of such devices is very great. I have in my possession 165 different forms of automatic sprinklers, most of which would not be reliably operative in case of fire, or they are so constructed as to be especially prone to leakage, or to disability by corrosion.

There are a great many forms of tests necessary to predicate the degree to which various automatic sprinklers will withstand these numerous elements tending to impair their efficiency. The paper in question deals with only one matter which must be considered; the other features being largely those relative to capability of withstanding water pressure, proper discharge of water, distribution of

water, freedom from leakage either by jarring or by water pressure, freedom from sticking at the valve, freedom from disability by corrosion, positive operation when the solder is heated without being stopped by water impinging on the solder joint, opening temperature, and lastly the question of relative sensitiveness.

An automatic sprinkler does not open at the fusion point of the solder, which is about 167° Fahr., but at the critical point of the alloy or the granular state, which is about 162°. When the soldered joint becomes heated to this temperature, it will yield, and the relative sensitiveness of action in the presence of a rapidly-increasing temperature is largely governed by the thickness of brass over the soldered joint.

The thin film of solder between sheets of brass is much more sensitive to rapid changes of temperature than a mercurial thermometer with its globular mass of mercury surrounded by a glass bulb, which is relatively a poor conductor of heat, and comparable to the lagging on a steam-pipe.

By reason of this relatively slow action of mercurial thermometers, they cannot give trustworthy results in measuring a rapidly-increasing temperature, unless the apparatus is so arranged that the rate of this increase can be controlled and made to conform to a constant rate, and the tardiness of the thermometer standardized to conform to this condition after the very ingenious manner set forth in the paper of Mr. Nagle.

I do not propose to discuss the methods of oven tests used by all sprinkler manufacturers. They have no pretence to scientific value, but they do serve a very useful commercial purpose in showing to prospective customers that an automatic sprinkler will operate promptly when exposed to slight heat.

In testing the opening of automatic sprinklers, I employ two entirely different processes, one being for the purpose of ascertaining the opening temperature of the sprinklers. The apparatus consists of an oven made out of a street-lamp, with a pipe entering at the top on which the sprinkler can be placed in a pendant position similar to that which it has when in use. A water pressure is obtained by connecting the pipe to a reservoir made out of a fire extinguisher, the upper part of which contains compressed air, the water being forced in at the bottom by a pump. In this manner the water pressure on the sprinklers can be controlled without shocks due to the action of the pump, and furthermore, when necessary it is possible to have a circulation of water in the

pipe in order to conform to the condition which always exists in the operation of any sprinkler in a line of pipe after the first one is opened.

The opening temperature of the sprinkler is tested by means of a water bath. A double pail of water is suspended within the oven, so that the soldered joint of the sprinkler will be in about the middle of the inner pail, and thermometers are inserted with their bulbs as near to the soldered joint as practicable. The water is slowly warmed by a gas flame until a temperature of about 145° is reached, when it is kept stationary for a while, and then increased at a rate of not more than 1° a minute until the sprinkler opens.

In this manner the temperature of the opening of the sprinkler under any desired pressure is very accurately measured, and the whole process can be clearly observed, as the glass sides of the lantern prevent the hot water from being thrown about when the sprinkler opens.

The same apparatus, with the exception of the water bath, is used to test the positiveness of the operation of sprinklers, not merely of those which have been corroded, but also such sprinklers as are liable to be fixed in the act of opening by the water issuing from the opening sprinkler and striking and cooling the soldered joint before the parts are free. This is a defect to which quite a number of sprinklers are liable at time of trial, but it is doubtful whether sprinklers will be found in that position after an actual fire, as the amount of issuing water would not be sufficient to quench the fire, and, on the other hand, the fire would naturally extend until it produced sufficient heat to warm the water in the pipes to the melting point of the solder, and in that manner boil open the sprinkler.

One sprinkler would open as fast as any other using the same solder, if the heat rose slowly enough; and, on the other hand, the ratio of speed of operation of sprinklers to each other is dependent upon the rate at which the temperature increases.

In regard to the sensitive tests of sprinklers, any measurement must be confessedly an artificial one, and so must the standard be as far as the method used is capable of exact reproduction. I have made a great many trials, both with gas flames and with solid fuel, but without obtaining satisfactory results. With gas flames, even when controlled by regulators, the calorific power of the gas would vary from day to day, and under the same condi-

tions of pressure the gas, from different causes, would never be uniform. Even the burning of measured amounts of alcohol or similar fuel did not give exact results, as the increment of temperature would be quite variable, because with the amount of alcohol used under a slow sprinkler the temperature of the burning mass would increase to such an extent as very materially to add to the rate of heating; and, therefore, the conditions of a slow sprinkler were not those of a sensitive one. At one time I tried to measure the sensitiveness of sprinklers by kindling fires from a measured amount of excelsior, in a building erected for the purpose, but the results were exceedingly unsatisfactory, varying in the time of operation from a ratio of two to one in consecutive experiments.

I finally used steam as the best means of obtaining a uniform increment of heat. The apparatus consisted of a large table, in the middle of which projected an open Parmalee automatic sprinkler, which is in form a reaction turbine, and was used for this purpose because it would throw the steam out radially over the surface of the table. This Parmalee sprinkler was connected with a large section of pipe underneath, which contained some water in the lower part of it. Steam was blown into the lower part directly against the water, when necessary, in order that the reservoir should be filled with steam free from superheating, and controlled by a valve to the pressure of 15 lbs. The sprinklers were placed above this table, each containing a small quantity of water, and by means of rods and weights subjected to an opening force equal to 15 lbs. per square inch. An electrical apparatus was so attached that when a sprinkler opened a bell corresponding to that sprinkler would ring in an adjoining room, and thus notify the assistant who was making a record of the time by means of a stop-watch.

When a test was to be made a box of 30 cubic feet capacity was lowered face downward upon 'this table, and the valve opened very slowly until the temperature of the box was raised to 100° Fahr., this being the temperature to which sprinklers are liable to be exposed continuously in the extremes of summer and winter weather. Then the steam-valve would be opened so that a pressure of 15 lbs. would be admitted to the steam-drum, and from thence through the sprinkler into the box.

The results of this method were very satisfactory, for the variation in the time of opening of sprinklers in the same lot was not

over one-fifteenth of the time required for their operation. The time required by automatic sprinklers of different makes, but all using solder of substantially the same fusion point, varied from 12 to 95 seconds. Under these conditions the various types of sensitive sprinklers opened in from 15 to 21 seconds, the earlier types of sealed sprinklers requiring varying amounts of time up to the 95 seconds referred to above.

Until this paper was presented to the Society, I have never heard of air measurements being used as a means of determining the opening temperature of automatic sprinklers, except in the commercial manner already mentioned, and I do not think that satisfactory results could be reached in that manner except in connection with a method of standardizing, first, the tardiness the thermometer in its relation to the uniform increment of heat, and, second, the smaller tardiness of the automatic sprinkler in respect to the same condition.

*Mr. Nagle.**—I have but little to say in reply to the discussion offered, for the subject of inquiry in my paper was limited to the scientific expression of the heat required to open automatic sprinklers.

Water is unsuitable as a means of applying the heat, for the reason given in the paper, since it contains several thousand times as much heat in the same volume as air, and hence is altogether too coarse a measure ; that is, it would not indicate with sufficient delicacy the different quantities of heat required. Steam would be worse than water, for, owing to its great elastic force, it would circulate so rapidly as to maintain the temperature uniform, and in such large quantities as to make the results vary but little. Mr. Woodbury's tests for sensitiveness show this to be true. He found that under the conditions of his apparatus "the various types of sensitive sprinklers opened in from 15 to 21 seconds." In my apparatus I tested two sprinklers, A and B, which, in construction, were not extreme forms of the sensitive type, in fact, apparently much alike, and yet I measured, as shown, 170 thermal minutes for the one and 205 for the other.

It seems to me, also, to be the better plan to keep within the conditions of practice in scientific investigations, so far as practicable.

A large oven has the disadvantage of allowing the air to circulate. It will have one temperature in one part of the oven, and

* Author's closure, under the Rules.

another temperature in another part. What we want is a volume of the same hot air about the little sprinkler, which is only a few inches in diameter, at each test; that is all, and I believe that can be obtained better in a small oven than in a large one.

Putting water into the sprinkler pipes when you test the sprinkler involves a practical difficulty. The moment you throw water into the oven you change the conditions for the next test, and it is practically impossible to restore the oven conditions after water has been thrown into it. Even if you throw the same quantity of water each time, and every care is taken to wipe it out alike each time, there is still enough moisture in uncertain quantity to affect the conditions of the next test.

CCCLXXXIV.

TESTS OF SEVERAL TYPES OF ENGINES, UNDER CONDITIONS FOUND IN ACTUAL PRACTICE.

BY R. C. CARPENTER, LANSING, MICH.

(Member of the Society.)

THE test was undertaken with the especial object in view of ascertaining what economy would be gained by substituting compound or compound condensing engines, running at comparative high speeds, for simple engines, performing the same class of work.

The engines under consideration consisted, first, of a simple engine, automatic governor, 14 inches diameter by 20-inch stroke. Second, a compound non-condensing engine, horizontal tandem type; high-pressure cylinder, 14-inches diameter by 20-inch stroke; low-pressure cylinder, 21 inches diameter by 20-inch stroke; automatic governor on high-pressure cylinder, fixed eccentric for low-pressure cylinder. Third, simple condensing engine, of the Reynolds-Corliss pattern, 14 inches diameter by 36-inch stroke. Fourth, compound condensing engine, horizontal tandem type, automatic governor connected to both valves; high-pressure cylinder, 9½ inches diameter by 20-inch stroke; the low-pressure cylinder, 21 inches diameter by 20-inch stroke. The condenser pump was worked by a belt from the main shaft of the engine; boilers fed by a separate piston pump. Fifth, two compound condensing engines, horizontal tandem type, automatic governor; high-pressure cylinders, 14 inches diameter by 20-inch stroke; low-pressure cylinders, 21 inches diameter by 20-inch stroke. Condenser and boiler feed-pump run with belt from main shaft of the engine, taking water from the hot well, in ordinary conditions; during the test, however, the water had to be taken from the city water-works. Sixth, two compound condensing engines, with throttling governor, and cut-off adjustable by link motion. The condenser was an independent pump condenser of the Hughes type. Feed-water for the boilers usually taken from the hot well, but during the test had to be taken from the city water-

works. Seventh, two triple expansion condensing engines, horizontal tandem type, with automatic governors. Condenser pump run by a belt to the main shaft of the engine; feed-water taken from hot well. In this case crude oil was used as a fuel.

In no case did the engines tested have steam-jackets. The engines were in ordinary working conditions, and in no case especially fitted up to be tested. In a few cases the valve motions were slightly deranged, and could not be adjusted before the tests were made, so that these results represent rather the actual work of the engines under the conditions found than the capacity of the same engines under the most favorable conditions. In some cases the duty no doubt approached nearly a maximum for the conditions under which it was tried.

The simple automatic engine shows a water consumption per I.H.P. less than could probably be obtained by the average engine of that class, while the compound non-condensing engine, on the other hand, shows a greater water consumption.

It had been the intention to repeat the tests on both the above engines under different circumstances, but time and opportunity have not permitted.

Methods of Testing.—The methods used in testing the engines were nearly the same in each case, and were as follows:

The different engines tested were located in the cities of Lansing, Jackson, Albion, Adrian, and Jonesville, Mich., and each surrounded with very different conditions, so that in many cases it was a matter of considerable difficulty to arrange them so as to make the proper measurements.

Measurement of Feed-water.—After the trial of several methods, with only partial success, at the first plant to be tested, the plan was adopted of using three barrels. In two of these barrels 400 pounds of water was weighed, having about the temperature of the feed-water. The line of this water surface was marked by driving nails, about four inches apart, completely around the barrel. These two barrels were then mounted over the third and provided with short pieces of two-inch pipes, furnished with valves, opening into the third barrel. This pipe was made very short, so that in case of any leakage it could readily be seen. A suction pipe leading to the boiler feed-pump was connected to the third barrel. An arrangement consisting of a swing pipe, or a hose, was used to fill the barrels alternately. The method of using was to fill one barrel while the other was emptying. This simple arrangement is one

which can easily be arranged, and proved satisfactory in all the cases tried. It has a capacity of about 150 pounds per minute.

In two cases water was obtained from tanks with considerable capacity, and although every precaution was taken to insure accuracy in measurement, yet more or less doubt has arisen respecting the results.

Care was taken that all the water supplied to the boiler during the test should be measured by the method described.

During the test the amount of water in the boiler was kept as nearly constant as possible, and at the close the height as shown by the gauge was brought to the same point which it held at the beginning of the test.

The Fuel.—The fuel used in Michigan is generally Ohio soft coal, and varies greatly in quality. In a few instances the coal was very wet, having been in recent snow-storms in open cars. The only way to secure uniform results was to make comparisons with combustible.

The method of making the fuel tests was to start with very low and clean fires, clean ash-pits, and clean boilers, and leave the fires in the same condition at the end of the test. In one case it was practicable to start with entirely fresh fires. The amount of fuel put into the furnace was in each case weighed; all the ash, cinders, and soot produced during the test were weighed and deducted. To obtain the moisture, 100 pounds of coal in its original condition was dried, and the loss was taken as the per cent. of correction to be made for moisture.

The fuel losses deducted gave us the combustible.

Calorimeter.—To determine the amount of entrained water in the steam, calorimeter determinations were made at intervals during the test. The ordinary barrel calorimeter was used, it being mounted on the best possible pair of scales to be obtained in each case. The thermometer used was a standard steam thermometer, 24 inches long, made by Henry J. Green & Co., of New York.

The method finally adopted was to add several charges of wet steam to the calorimeter, taking the temperature at each interval, before emptying the water and filling again. I find that a very efficient mixer is made by discharging the steam near the bottom of the barrel through a tee placed crosswise of the pipe, as shown in the sketch. If the pipe is left in the barrel an air-cock or vent should be provided to prevent the water rising in the pipe by condensation of steam. I found that stirring did not change the

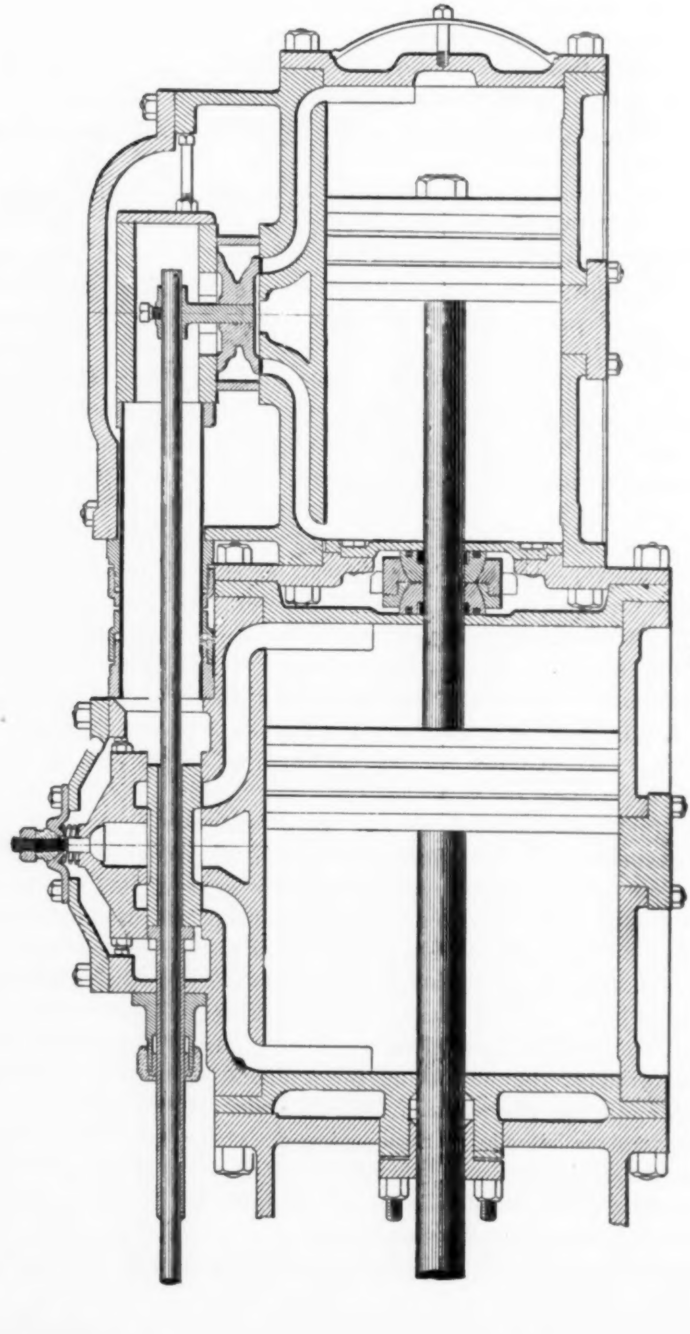


Fig. 115.

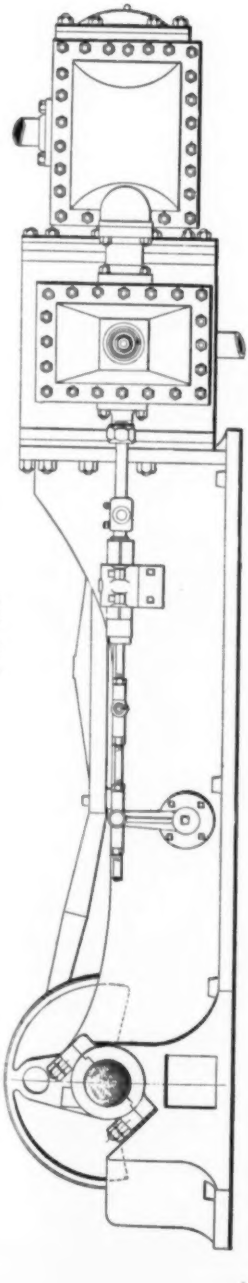


FIG. 116.

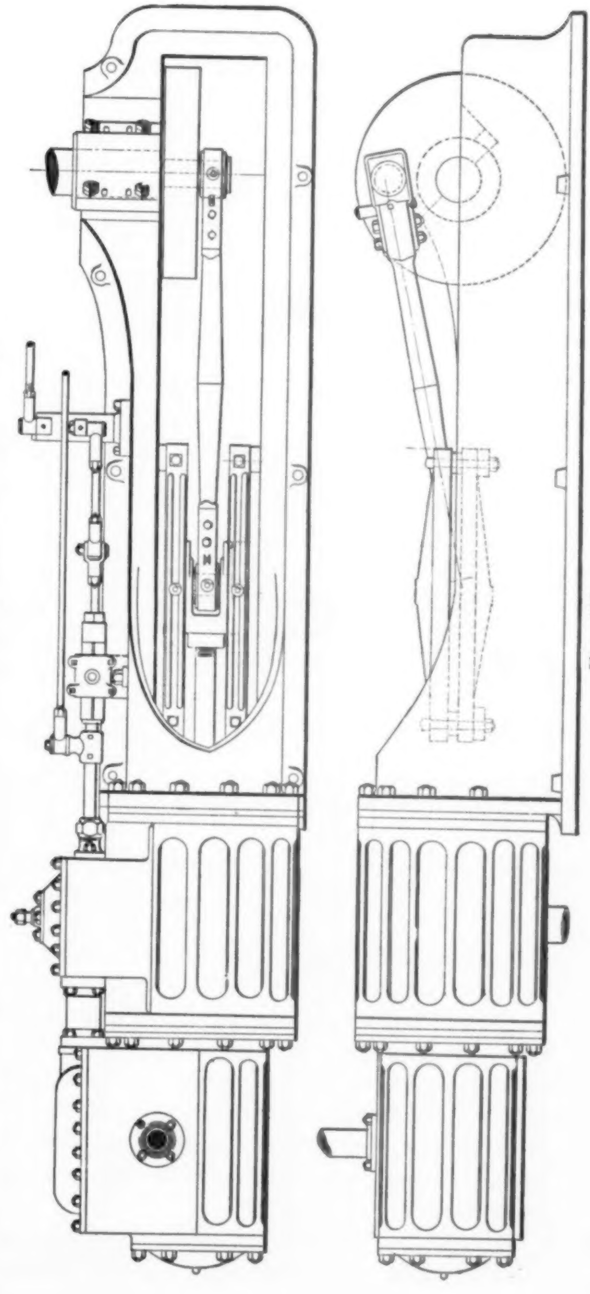


FIG. 117.

temperature in the least, if readings were taken immediately after the steam was discharged. As the best scales to be obtained did not read closer than one-fourth pound, I adopted the expedient of fixing the poise at a given place and of adding steam until the weight was balanced.

Indicating.—Two or three indicators were used as required. In each case these were set opposite the center of the cylinder, and generally connected to the ends with a three way cock. I tested

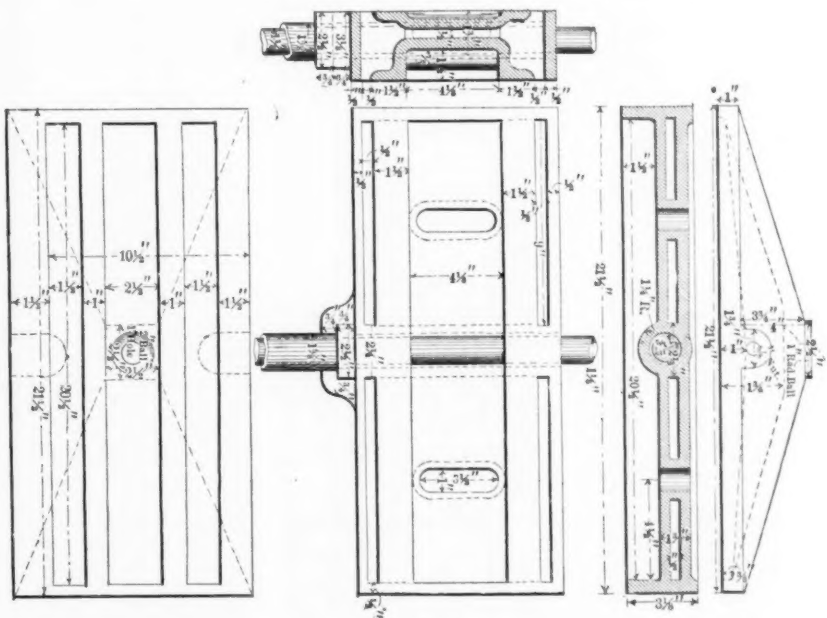


FIG. 118.

the principal indicator springs used, under steam and against a mercury column, in the mechanical laboratory of Sibley College, of Cornell University, through the kindness of Prof. R. H. Thurston, and found no error exceeding one-half of one per cent.

Several kinds of reducing motions were tried, but in most cases a pendulum fastened over the center of the guides, and provided with arcs at the proper distance from the center of motion, was used. In one case, when the motion was slow, the lazy tongs were used.

Description of the Engines.—The compound engines subjected to the following tests were built by the Lansing Iron and Engine

Works, of Lansing, Mich., from the designs of Mr. S. E. Jarvis. The engines are all of the horizontal tandem type, furnished with automatic cut-off in most cases. The first engine of this design was built in 1888. The drawings submitted show very clearly the peculiar construction of the valve chests and the arrangement of the cylinders. The wood-cuts show the general appearance of the engines when finished.

The valve gives exceptionally quick opening, having passages cored out so as to take steam from four directions simultaneously. The valve is shown in the accompanying drawings (Figs. 115, 116, 117, 118).

The following pages give details of the various tests referred to:

TEST OF AUTOMATIC SIMPLE ENGINE AT JONESVILLE, MICH.

Size of engine, 14″ diameter, 20″ stroke. This engine was built by the Lansing Iron and Engine Works, and at the time of making the test it had been running one year. The valves were not properly set at the time of making the test; otherwise the engine was in good condition.

No.	Rev.	Steam Pres.	M.E.P.		I.H.P.	
			Hd.	Cr.		
1	173	80	22.1	12 2	45.08	Indicator cards taken with 50 spring. (Fig. 118)
2	174	80	22.5	10.9	45.00	
3	173	80	23.5	14.4	52.30	
4	173	75	23.1	12.3	47.10	
5	166	66	25.5	14.6	48.30	
6	166	67	21.9	12.5	41.80	
7	166	67	22.5	13.7	46.71	
8	171	70	22.5	12.5	46.02	
9	172	70	20.5	11.1	42.45	
10	168	68	19.8	9.2	35.23	
11	174	70	21.2	12.4	45.90	
12	172	68	21.9	12.5	45.35	
13	176	70	21.8	12.5	47.37	
14	176	68	21.6	12.5	46.81	
15	162	65	21.5	12.5	45.82	
16	176	70	21.5	11 9	45.71	
17	168	63	20.2	12.2	43 21	
18	166	65	21.4	11.3	44.82	
19	166	70	22.1	12.0	44.02	
20	168	73	21.5	10.3	41.57	
Average...	171.5	70.25	21.7	12.27	45.50	

The average measurements of the indicator cards gives us admission pressure in cylinder of 63.3 lbs.; release pressure, 8.2 lbs.; back pressure, about one-fourth pound above atmosphere.

The steam per I.H.P. computed at end of stroke from the card was 24.53 lbs.

FUEL.

The fuel used in this test was Jackson Hill lump coal, the average consumption per hour being 242.9 lbs.; per I.H.P., 5.34 lbs.

The ash and clinker at end of test amounted to 73 lbs., or 18 lbs. per hour, the loss from moisture obtained from drying the coal amounting to .06 lbs., making the loss or waste of the coal .112%. This would make the combustible per I.H P. 4.76 lbs.

WATER USED.

The water used in this test was obtained from a tank 35 inches in diameter, which was alternately filled and emptied through a distance of 3 feet in height. The temperature of the feed-water was 43.4° Fahr., the amount used per hour being 1,362 lbs. by calculation.

The water used per I.H.P. was 30.9 lbs. Duration of test, 4 hours 33 minutes.

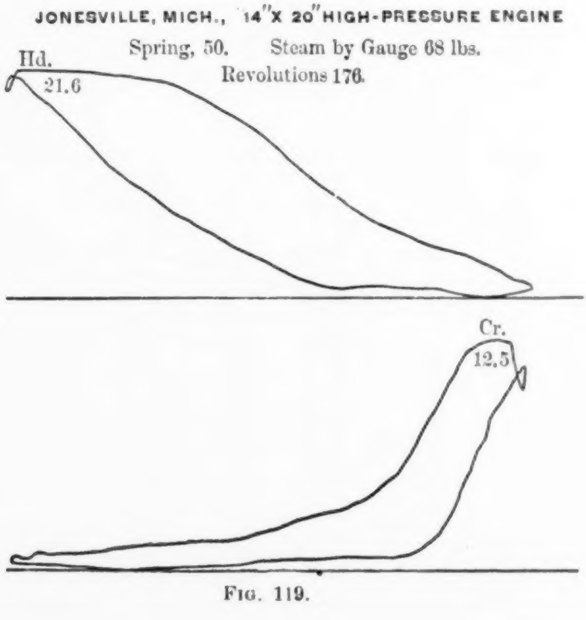


FIG. 119.

EVAPORATION.

The evaporation per pound of coal from and at 212° was 6.7 lbs. of water; the evaporation per pound of combustible, 7.49 lbs. of water.

BOILER.

The boiler, of ordinary tubular pattern, 5 feet diameter by 14 feet in length.

TEST OF COMPOUND NON-CONDENSING ENGINE AT LANSING WHEEL WORKS, LANSING, MICH.

High-pressure cylinder, 14″ dia., 20″ stroke; low-pressure cylinder, 21″ dia. by 20″ stroke; built by the Lansing Iron and Engine Works. The exhaust steam from this engine is used to heat the shops and dry kilns; the gauge placed on the exhaust pipe showing constantly from 3 to 6 lbs. back-pressure. A test was first made Dec. 1, 1889, of three hours' duration; some doubts arising respecting the water measurement, the test was repeated, Dec. 23d. The following are the results:

No.	Rev.	Steam Pres.	High Pressure.			Low Pressure.		
			M.E.P.		I.H.P.	M.E.P		I.H.P.
			Hd.	Cr.		Hd.	Cr.	
1	152	83	19.5	20.7	47.51	3.1	4.5	19.77
2	154	80	17.6	19.8	43.60	3.9	4.9	23.72
3	156	80	19.2	19.5	46.94	3.7	4.6	26.48
4	156	83	18.8	20.4	47.06	4.6	5.1	19.66
5	155	80	16.8	18.0	41.93	3.4	3.8	22.12
6	154	78	17.6	18.9	43.72	3.8	4.4	19.57
7	151	77	18.3	20.3	45.32	3.5	4.1	24.00
8	152	80	19.5	20.7	47.51	4.5	4.5	22.66
9	154	81	18.9	20.7	47.43	4.3	4.3	23.18
verage.	......	....	....	....	45.55	....	....	22.35

Total .... 67.9 I.H.P.

During the test water was fed to the boiler by injector, which probably consumed at least .... 2.00 "

Making the total power .... 69.9 "

The water used was drawn from a tank, and weighed 2,418 lbs. per hour, by calculation; consumption of water per I.H.P. equals

345 lbs.; the consumption of steam per I.H.P. equals 32.8 lbs., an amount 5% less.

FUEL.

On beginning the test fresh fires were started with steam raised. The consumption was 365 lbs. of coal, or 298 lbs. of combustible per hour; coal per I.H.P. equals 5.2 lbs.; combustible per I.H.P. equals 4.2 lbs.

EVAPORATION.

Temperature of feed-water, $39\frac{1}{2}°$ Fahr.; 6.3 lbs. of water heated from $39\frac{1}{2}°$ Fahr. and converted into steam at a pressure of 83 lbs. per pound of coal, equivalent to 7.8 lbs. per pound of combustible; water evaporated from and at 212° per pound of coal, 7.4 lbs.; per pound of combustible, 9.2 lbs.

From the indicator cards (Fig. 119) the back-pressure in the low-pressure cylinder was 2.5 lbs. The water rate for the test, computed at end of stroke, is 25.8 lbs. per I.H.P.

SECOND TEST OF COMPOUND NON-CONDENSING ENGINE, DEC. 23, 1889, AT LANSING WHEEL WORKS, LANSING, MICH.

No.	Rev.	Steam Pres.	High Pressure, 14 x 20.			Low Pressure, 21 x 20.		
			M.E.P.		I.H.P.	M.E.P.		I.H.P.
			Hd.	Cr.		Hd.	Cr.	
1	150	83	17.6	21.0	45.02	4.4	5.3	26.02
2	148	80	18.7	19.5	42.97	5.8	5.9	30.53
3	147	80	17.1	18.6	40.83	3.5	5.2	22.62
4	146	80	18.0	20.1	43.46	3.5	4.8	21.23
5	140	70	17.4	18.0	38.54	3.5	4.3	19.10
6	145	80	17.5	19.5	42.00	3.3	4.5	19.36
Average.	......	....	....	....	42.13	....	....	24.61

Total 66.74
Add work of injector 2.00

68.74 I.H.P.

Water consumed—by calculation from measurement taken of tank—per hour, 23.61 lbs., or 34.3 lbs. per I.H.P. This, reduced by calorimeter measurements 5% on entrained water, gives the steam consumption per I.H.P. as 32.6 lbs.

FUEL.

The coal burned per hour was 342 lbs., or 5.12 lbs. per I.H.P.; the combustible per I.H.P., 4.01. Evaporation temperature of feed-

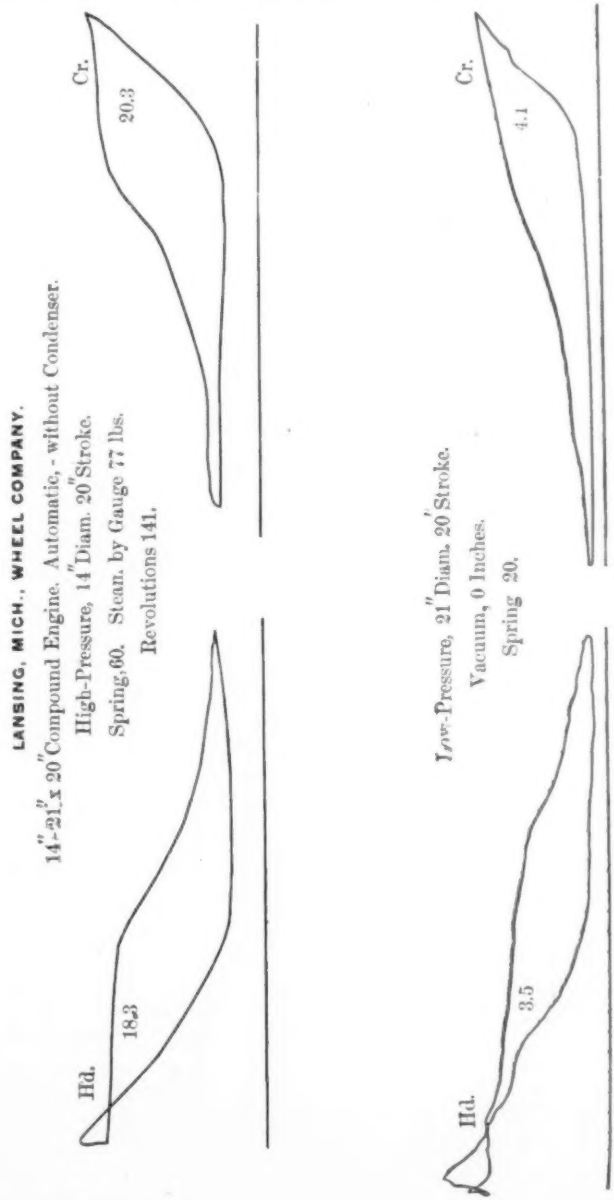


FIG. 120.

water, 39½° Fahr.; 6.3 lbs. of water heated from 39° Fahr. and converted into steam at 80 lbs. pressure per pound of coal, equivalent to 81. lbs. per pound of combustible; the evaporation from and at 212° Fahr. would be, per pound of coal, 7.46 lbs. of water, and per pound of combustible, 9.56 lbs. of water.

TEST OF A REYNOLDS-CORLISS CONDENSING ENGINE AT THOMAN'S FLOUR MILL, LANSING, MICH.

Size of engine, 14″ dia., 36″ stroke, with condenser. This engine had been in operation three years; was in good condition, as shown by the indicator cards. The pipes leading to this engine are all covered, and every precaution taken to prevent waste of steam. There were no leaks of any kind. The water for the test was pumped into a tank above the boiler from the hot well, with a temperature of about 100°, before the test began, from which it ran to the weighing apparatus, which was arranged as previously described.

The coal used during the test was wet bituminous pea coal, which was found to contain, by drying, 13% of moisture, and by weighing back refuse .09 of cinders and ashes.

Test started at 8.35 P.M. with fresh fires, clean ash-pits and boilers, and ended at 12.35 P.M. All refuse was weighed back.

INDICATOR CARDS.

No.	Time.	Rev.	Steam Pres.	Vacuum, Inches.	Feed-water Temp.	M.E.P.		I.H.P.		Total.
						Hd.	Cr.	Hd.	Cr.	
1	8.35	95	76	27¼	109	25.0	17.7	33.25	20.6	58.8
2	8.46	94	68	27	116	25.5	22.7	35.50	27.3	60.8
3	......	95	70	27	120	21.4	25.1	28.5	30.6	59.1
4	......	95	58	26½	120	23.0	26.6	30.55	32.47	62.9
5	9.30	95	59	27	128	26.4	24.5	35.12	29.86	65.0
6	......	96	67	27	118	22.0	25.6	28.17	31.58	59.7
6 +	9.40	95	60	27	120	23.5	24.5	31.3	29.8	61.2
7	10.7	95	70	27	130	24.2	24.2	32.3	29.7	61.9
8	10.30	95	70	27	118	21.	25.7	27.9	31.0	58.9
9	10.40	94	70	26¾	118	21.	24.2	27.6	29.3	56.9
10	10.50	95	68	27	118	25.	25.5	33.2	31.6	64.8
11	11.00	95	68	26½	128	22.	26.6	29.4	35.7	65.1
12	11.10	95	70	27	119	25.7	26.0	34.2	30.9	65.1
13	11.30	95	60	27	130	24.2	23.0	32.2	28.1	60.3
14	12.10	96	68	26½	125	27.5	24.0	36.9	29.3	66.2
15	12.20	96	68	26½	125	28.0	24.5	37.6	29.8	67.4
16	12.15	96	70	27	125	23.5	21.2	31.6	26.2	57.9
17	12.35	96	65	27	125	26.0	27.4	35.0	33.8	68.8
Average			67.4	26.88	121.6	....	....	....	....	61.99

WATER TEST REYNOLDS-CORLISS ENGINE.

Started and ended test with water at same point in boiler.

Water used, 7,112 lbs. in 4 hours.
" " 1,778 lbs. per hour.
28.67 per I.H.P.

CALORIMETER.

Calorimeter test showed steam to contain 6.7% of water; deducting this from above leaves steam per I.H.P. per hour 26.75 lbs. (Fig. 120).

CALORIMETER MEASUREMENTS.

Steam Pressure.	W Wt. Condensing Water.	w Wt. Wet Steam.	T Temp. Condensing Water.	T' Resultant Temp.	H Total Heat in Dry Steam.	$H - T'$	$\frac{W}{w}$	$T' - T$	I Latent Heat.	$\frac{x}{w}$ % Moisture.
71	300	15	61.3	112.6	1178.6	1066	20	51.3	890.6	.046
66	300	20	61.3	128.4	1177.3	1048.9	15	67.1	893.6	.048
67	300	15	63.2	107.6	1177.6	1070.0	20	44.4	893.	.092
67	300	20	63.2	124.4	1177.6	1053.2	15	61.2	893.	.083
								Average......		.067

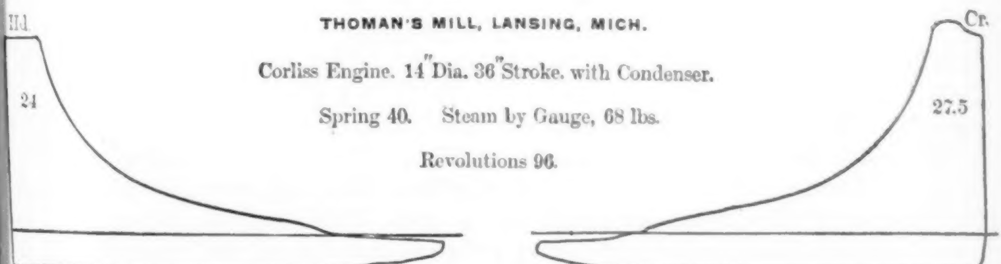


FIG. 121.

This water was heated during the test about 21° by steam directly from the boiler.

COAL TEST.

Burned in 4 hours 1,135 lbs., or 283.75 lbs. per hour. This is equal to 4.63 per I.H.P. Weighed back cinders and ashes, 103 lbs., equal to .0908%. Dried 100 lbs. of coal, and found 13% loss.

Combustible burned during test, 898 lbs.
" " per hour, 224.50 lbs.
" " " I.H.P., 3.61 lbs.
" " " I.H.P., 3.58 lbs., had feed-water tem. 110°.
Coal burned per hour per I.H.P., 4.59 lbs., had feed-water tem. 110°.

FLOUR MANUFACTURED.

During the test 5.79 barrels were manufactured per hour, making the consumption of coal equal to 49 lbs. to a barrel of flour, or of combustible 38.52 lbs. to the barrel of flour.

TEST OF BOILERS, CORRECTED FOR WORK OF HEATER.

Steam used per I.H.P., 26.87.
Evaporation from 212° per lb. of combustible, 8.29 lbs.
" " " " coal, 6.82 lbs.

The boilers in the above plant consist of two, each 4 feet diameter by 12 feet in length, and have been in use a long time. The middle vertical row of tubes has been omitted because of a theory of the owner that they retarded the upward circulation of the water.

COMPOUND CONDENSING ENGINE, ALBION, MICH., ELECTRIC LIGHT STATION, BUILT BY LANSING IRON AND ENGINE WORKS.

This engine has cylinders 9½-inch diameter and 21-inch diameter, 20-inch stroke, arranged as a horizontal tandem. The engine was constructed with a fixed eccentric for the low-pressure cylinder, and with a movable eccentric, automatic governor, for the high-pressure cylinder.

The engineer had connected both valve stems together and attached them to the automatic governor. The test was continued throughout four hours, which was as long a time as it was possible to obtain without great variation in load. The water was measured throughout the test, being pumped from the hot well into a barrel holding 300 lbs.; thence it was run into a second barrel, from which it was pumped by a donkey feed-pump.

The temperature of the feed-water was 108°. The power consumed by the donkey pump was estimated by applying a brake to the drive-wheel while under steam. The average of the results indicated 2.6 H.P. The average results during the test showed 95.47 I.H.P. as the work of the engine, making the total work of the plant 98.07 H.P.

The average calorimeter results showed 5.6 water entrained in the steam. The steam for the calorimeter was taken from the main steam-pipe about 3 feet from the cylinder, and was doubtless a fair sample of that supplied the engine. The steam-pipes were not covered, and the moisture is no doubt due to condensation in pipe.

INDICATOR CARDS, TANDEM COMPOUND, 9½ AND 21 x 20.

No.	M.E.P.		I.H.P.		Steam Pres.	Vacuum by Gauge.	Rev.
	High.	Low.	High.	Low.			
1	51.0	7.55	57.70	41.75	130	26	158
2	50.6	7.25	57.24	39.60	130	26	158
3	50.48	6.00	57.13	33.60	130	26½	158
4	50.15	6.40	57.46	35.84	130	26⅓	160
5	50.45	7.50	57.79	42 00	133	26½	160
6	49.5	7.10	56 75	39.76	130	26	160
7	49.6	7.25	56.83	40.60	132	26½	160
8	49.65	8.10	56.52	45.28	125	26	159
9	49.86	6 70	56.35	37.06	125	26⅓	158
10	48.75	6.90	56.30	39.92	134	26⅕	158
11	48.60	7.40	55.66	41.44	128	26$\frac{7}{16}$	160
12	45.45	6.85	52.70	38.84	132	26.6	162
13	42.75	6.75	50.60	38.75	135	26.6	162
14		6.50		37.50	130	26.6	162
			56.04	39.43			

Total I.H.P. engine	95.47
Work of pump	2.60
Total	98.07

Average speed, 160 revolutions.

Variation in speed, .02½%.

Variation in load, .06½%.

FUEL.

Fuel used during the test was very wet slack coal; 1,000 lbs. burned 3 hours, 22 minutes, or at the rate of 296$\frac{7}{10}$ lbs. per hour, or 3.05 lbs. per I.H.P. The combustible used was found by drying 100 lbs. of the coal, and also by weighing back the ashes and clinkers. The result showed the following waste per 1,000 lbs.: moisture, 132 lbs.; ash and clinkers, 226 lbs.; net combustible, .647% of coal.

Combustible per I.H.P., 1.97 lbs.

WATER USED.

The water used per hour was 1,950 lbs. The dry steam was .056% less 1,841 lbs. per hour. The consumption of steam per I.H.P. 18.77 lbs.

ALBION, MICH., ELECTRIC LIGHT STATION.
9½"—21" x 20" Compound Engine—Automatic.

High-Pressure, 9½" Dia., x 20" Stroke.
Spring, 60. Steam by Gauge 130 lbs.
Revolutions 160.

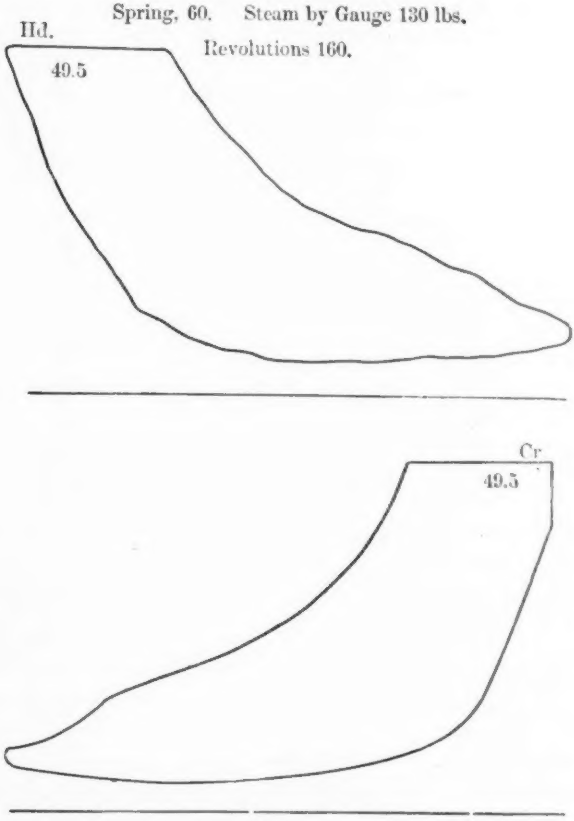


FIG. 122.

The boilers in use at this plant are two tubular boilers, each 5 feet diameter and 14 feet long, placed one above the other and connected by several vertical pipes. The fire is enclosed in the lower boiler, as in the marine boilers. The special design was adopted because of its great strength. It has also proved to be a very economical boiler, as shown by the results, and furnishes a good quality of steam. The feed-water furnished the boiler was at an average temperature of 102°, and the steam used was raised to 130 lbs. The evaporation from 212° per pound of coal was 6.09 lbs. of water; per pound of combustible, was 10.66 lbs. of water;

the evaporation from 102° per pound of coal was 6.12 lbs.; the evaporation from 102° per pound of combustible, 9.46 lbs.

From the indicator cards (Figs. 121 and 122) we find the average

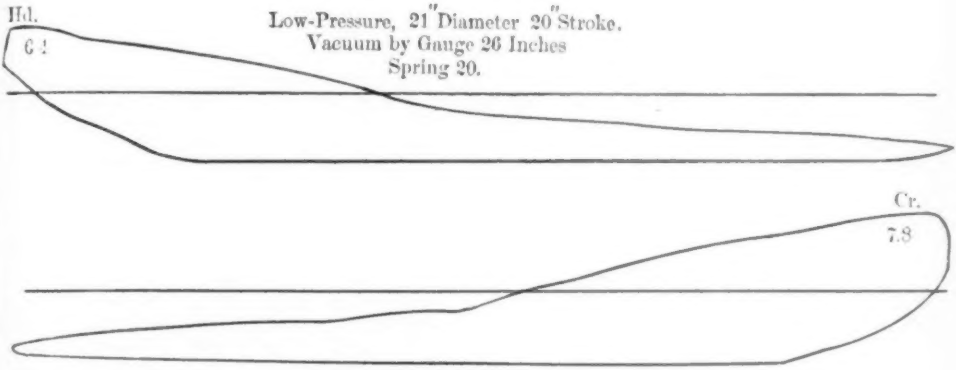


FIG. 123.

admission pressure in high-pressure cylinder was 113.3 lbs.; in low-pressure, 7.3 lbs. above atmosphere; the back-pressure in the high-pressure cylinder 11 lbs. above, and in the low-pressure cylinder 8 lbs. below the atmosphere. The water consumption measured from the indicator card at the end of stroke equalled 16.04 lbs. per I.H.P.

TEST OF TWO COMPOUND ENGINES, LANSING ELECTRIC LIGHT STATION, LANSING, MICH., BUILT BY LANSING IRON AND ENGINE WORKS.

These engines were of the tandem type, each with high-pressure cylinder 14-inch diameter by 20-inch stroke, and low-pressure cylinder 21-inch diameter by 20-inch stroke. Steam was supplied by two boilers of the ordinary tubular type, 5 feet diameter by 14 feet long. The whole plant at the time of testing was new, having been in use about 18 days. The steam-pipes were, however, of considerable length and uncovered. The method of testing was substantially as described, and it is deemed sufficient in this place to simply give the results. The two engines were coupled together, and as far as possible diagrams were taken simultaneously.

WEST ENGINE.

No.	Rev.	Steam Pres.	Vacuum by Gauge.	High-Pressure, 14 x 20.			Low-Pressure, 21 x 20.		
				M.E.P.		I.H.P.	M.E.P.		I.H.P.
				Hd.	Cr.		Hd.	Cr.	
1	170	100	23	15.8	20.0	47.33	7.8	7.3	44.92
2	170	100	23	16.1	22.3	50.77	8.5	8.1	45.10
3	173	98	23¼	5.8	16.7	27.99	8.5	9.0	52.97
4	171	100	23	15.9	22.3	50.52	5.9	7.0	38.62
5	171	100	23	13.5	22.0	47.13	8.4	7.9	49.39
6	170	100	23	14.0	21.3	46.67	8.2	8.1	48.49
7	172	100	23	11.5	21.3	43.87	7.4	7.3	44.98
8	170	100	23	13.8	20.5	47.19	7.9	8.3	48.22
9	175	100	23¼	3.1	15.6	24.07	5.8	5.8	36.53
10	174	100	23	16.0	31.5	63.04	13.0	13.3	79.62
11	174	98	23¼	25.0	28.5	71.6	13.2	12.8	78.26
12	174	100	23	3.1	14.0	23.29	3.7	4.5	25.33
13	178	105	23	3.0	14.0	23.71	4.3	5.0	27.00
14	173	100	23	10.3	18.75	39.35	7.4	7.5	45.54
Average.	172.4	160	23.06	11.9	20.6	43.30	7.93	8.8	47.50

Total I.H.P. 90.8

EAST ENGINE.

No.	Rev.	Steam Pres.	Vacuum by Gauge.	High-Pressure, 14 x 20.			Low-Pressure, 21 x 20.		
				M.E.P.		I.H.P.	M.E.P.		I.H.P.
				Hd.	Cr.		Hd.	Cr.	
1	168	100	23	25.0	26.5	67 26	10.9	12.3	68.21
2	170	100	23	23.6	27.0	66.81	11 8	11.2	69.06
3	167	100	23	22.2	25.2	61.97	11.1	11.2	65.56
4	170	100	23	24.5	28.1	70.03	10.5	11.2	64.56
5	170	100	23	23.5	27.0	62.21	10.7	10.9	64.56
6		97		23.8	26.5	66.60	11 2	11.4	68.13
7	168	100	23	23.4	26.7	65.44	10.7	11.2	66.45
8	168	100	23	22.5	25.0	62.04	11.0	10.5	64.68
9	171	100	23	22.5	26.0	65.22	11 3	10.7	64.38
10	172	100	23	24.5	26.3	67.94	12.5	12.5	66.22
11	173	100	23	24.3	27.0	69.30	12.7	12.5	76.28
12	179	100	23	25.9	32.5	80.87	12.7	12.5	78.28
13	180	100	23	26.5	32.3	82.29	13.3	12.5	81.16
Average.	172.5	100	23	24.0	25.8	68.28	11.4	11.5	69.04

Total I.H.P 137.32

From the indicator cards (Figs. 123, 124) for high-pressure cylinders the average admission pressure is 84 lbs.; the release pressure, 22 lbs.; the back-pressure, 14 lbs., reckoned from atmospheric

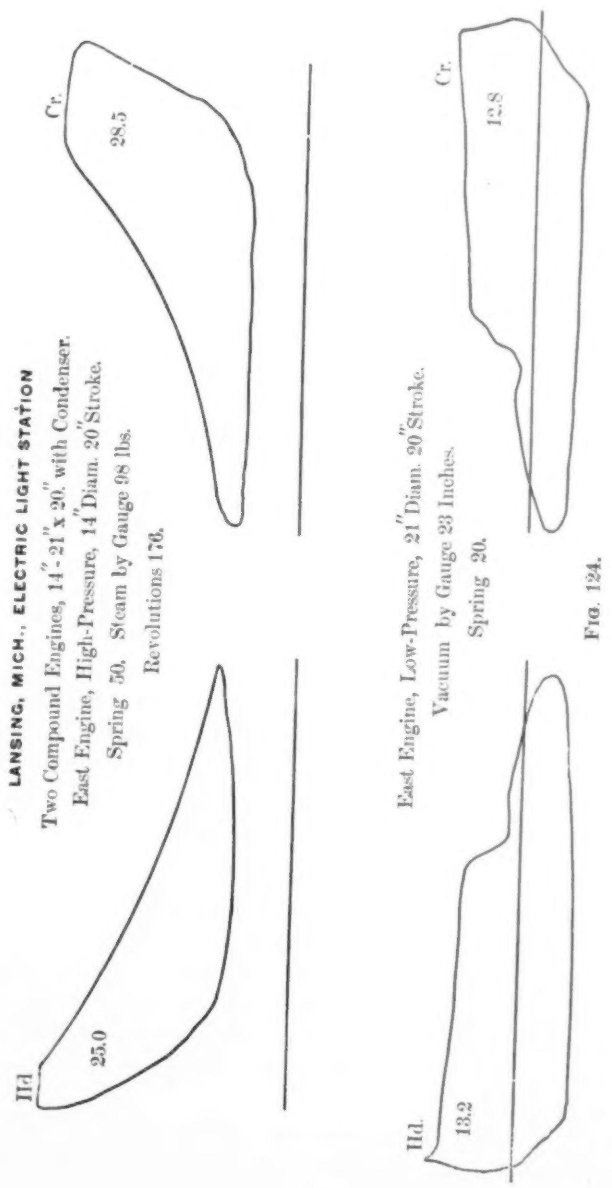

LANSING, MICH., ELECTRIC LIGHT STATION
Two Compound Engines, 14"-21" x 20" with Condenser.
East Engine, High-Pressure, 14" Diam. 20" Stroke.
Spring 50. Steam by Gauge 98 lbs.
Revolutions 176.
Hd.
25.0
Cr.
28.5
East Engine, Low-Pressure, 21" Diam. 20" Stroke.
Vacuum by Gauge 23 Inches.
Spring 20.
Hd.
13.2
Cr.
12.8

Fig. 124.

line. For low-pressure cylinder, estimated from same points, admission pressure is 10.1 lbs.; the release = 3.7 lbs.; the back-pressure = 7.5 lbs. The steam consumption per I.H.P. from cards at end of stroke is 15.84 lbs.

Total I.H.P. for both engines, 228.12.

FUEL.

Duration of test, 4 hours, 20 minutes. Weighed out at beginning of test, 2,661 lbs.; at end of test weighed back as unburned, 579 lbs.; weighed ash and clinkers, 194 lbs., or 9.3% of the fuel burned. The amount burned consisted of 700 lbs. of coke and 1,382 lbs. of coal.

To ascertain the condition, 100 lbs. of each fuel was dried, giving a loss for the coal of .018%, and for the coke, .05%. Making corrections for these losses, the fuel per hour was 475 lbs., of which 411.8 lbs. was combustible. This gives 2.08 lbs. fuel per I.H.P. per hour, or 1.80 lbs. combustible. During the test water was taken from the city water works, and had a temperature of 49° Fahr. Had the water been taken from the hot well at 110° Fahr., the consumption should have been as follows:

1.96 lbs. coal per I.H.P.

1.69 lbs. combustible per I.H.P.

WATER.

The water was weighed during the entire test, the amount consumed being 4,246 lbs. per hour. This is equal to 18.61 lbs. per I.H.P. Calorimeter tests were taken throughout the time of making the test, and showed 11% of entrained water; making this correction, the steam per I.H.P. was reduced to 16.57 lbs.

EVAPORATION.

One pound of combustible evaporated 9.983 lbs. of water, and raised it from an initial temperature of 49° Fahr. to the temperature of steam at 100 lbs. pressure. This is equivalent to evaporating 11.74 lbs. from and at 212°. Reducing this for the entrained water, 11%, gives us for the evaporation of 1 lb. of combustible 10.45 lbs. of water, and of 1 lb. of coal, 9.06 lbs. of water, from and at a temperature of 212° Fahr.

The boilers were of the ordinary tubular type, each 5 feet diameter by 14 feet long; they were new, clean, and the settings were in best of condition.

The details of the calorimeter test are given on p. 744.

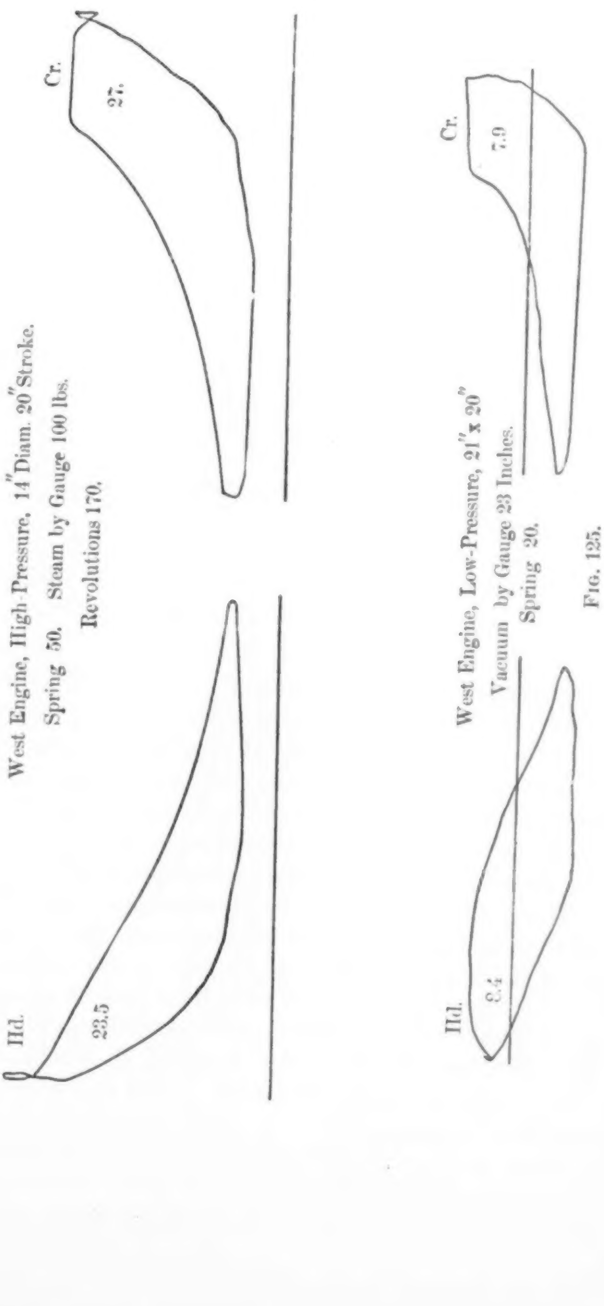

West Engine, High-Pressure, 14" Diam. 20" Stroke.
Spring 50. Steam by Gauge 100 lbs.
Revolutions 170.
Hd.
23.5
Cr.
27.
West Engine, Low-Pressure, 21" x 20"
Vacuum by Gauge 23 Inches.
Spring 20.
Hd.
8.4
Cr.
7.9

FIG. 125.

CALORIMETER TEST.

Weight of barrel, 61¼ lbs.

Took steam from main steam-pipe, near pump and at about same distance from the boilers as the engines. The pipes leading to the engines are about 20 feet long and uncovered.

CALORIMETER MEASUREMENTS.

Steam Pressure.	W Weight Condensing Water.	w Weight Wet Steam added.	T Temperature of Cold Water.	T' Resultant Temperature.	Total Heat in Dry Steam.	$H-T'$	$\frac{W}{w}$	$T'-T$	l	$\frac{x}{w}$ % Moisture.
100	290	10	50.8	83.4	1185.0	1001.6	29	32.6	875.4	.06
100	290	27.5	50.8	138.8	1185.0	1046.2	10.5	88	875.4	.13
100	327.5	10	55.8	85.8	1185.0	1099.2	32.75	30	875.4	.13
100	327.5	15	55.8	100.2	1185.0	1084.8	21.83	44.4	875.4	.13
100	332.0	10	99.2	125.6	1185.0	959.4	33.2	26.4	875.4	.09
100	332.0	15¼	99.2	139.2	1185.0	945.8	21.1	40.0	875.4	.10
100	315.0	15	56.2	102.8	1185.0	1082.2	21.0	46.6	875.4	.11
100	315.0	25	56.2	130.0	1185.0	1055.0	12.6	73.8	875.4	.13

TEST OF TWO COMPOUND CONDENSING ENGINES AT ADRIAN, MICH.

December 4, 5 *and* 6, 1889.

Both engines of same size—each 14 and 21 x 20″. These engines are furnished with throttling governors of the Waters pattern, two eccentrics and linked valve gear for adjusting the cut-off, an independent pump condenser of the Hughes pattern. This condenser maintained a vacuum of 27″ by the condenser gauge quite constantly. The indicator cards show an average vacuum of about 16½″ in the cylinders. Holes were drilled in each head of the condenser pump, and indicators attached and cards taken from both ends. These cards are shown by the sample submitted, and are quite phenomenal in form. This is in large part due to the peculiar motion of the piston, which stopped still in the middle of each stroke. The steam pressure on the piston rose to a maximum at this point, which was maintained throughout the stroke. Allowing 10% for clearance, the water rates, as shown by the cards from the condenser pump, computed at the end of the stroke, averaged 79.2 lbs. per I.H.P.

The water used by the condenser was taken by suction from the river, a distance of about 600 feet horizontally and 14 feet vertically. The peculiar action of the pump indicated that most of the pump-

ing was done on the latter half of the stroke. The following are the results obtained from the indicator cards:

HUGHES INDEPENDENT CONDENSER PUMP—10″ DIA., 12″ STROKE.

No.	M.E.P.		I.H.P.	Rev.	Absolute Release Pres.		Steam from Card per I.H.P.	
	Hd.	Cr.			Hd.	Cr.	Hd.	Cr.
5	20	20.4	2.60	25	42	44	75.5	74.3
6	16.7	19.1	2.12	25				
7	19.5	18.7	2.23	25	42.5	45	77.7	79.0
8	18.5	19.9	2.25	25	45.0	44.5	83.4	78.5
9	18.5	18.0	2.12	25	43.	45.	81.7	83.9
10	18.4	20.0	2.26	25	42.	45.	81.3	76.8
							79.9	78.5
Average							79.2	

THE ENGINES.

The north engine was run during the daytime to furnish motive power for an electric railroad on the Ray-Fisher system. Both engines were run at night for the electric lighting of the city, and in addition furnished the power needed by the street railway. Indicator cards were taken in the daytime to ascertain the power consumed by the electric road, but the complete tests, during which time the water and fuel were weighed, were only made at night.

COMPOUND ENGINE, ADRIAN ELECTRIC LIGHT STATION—NORTH ENGINE.

High-pressure cylinder 14 x 20, indicator spring 40.

Low-pressure cylinder 21 x 20, indicator spring 20.

Throttling governor. Hughes independent condenser. Electric street railway.

Time.	Amperes on Gauge.	Boiler Pres.	Vacuum by Gauge.	Rev.	I.H.P.		Total.
					High Pres.	Low Pres.	
2.35	40	80	27½	170	21.01	11.9	32.91
2.54	0	80	27½	170	4.3	6.24	10.90
3.05	40	85	27½	174	18.54	14.34	32.79
3.07	60	85	27½	172	22.46	16.82	39.39
3.28	40	92	28	166	20.15	12.04	32.19
3.40	0	83	27½	168	4.37	6.55	10.92
4.03	40	87	27	173	19.22	12.41	31.63

Average, 10.91 I.H.P. for 0 amperes.
" 32.38 " 40 "
" 39.39 " 60 "

During the same time indicator cards were taken on condenser pump, and these showed an average of 1.98 I.H.P. As this pump used 3.6 as much steam for the same work as the engine, this would be equivalent to 7.13 I.H.P. for engine. The average work of the street railroad, with two cars, required 40 amperes. Making the substitutions and reducing as shown before, we have:

Friction of north engine main line shaft and dynamos, 18.04 I.H.P.; average work for station, one engine and two cars, 39.15 I.H.P.; average work to run two cars without engine or dynamo, 21.11 I.H.P., or 10.55 H.P. per car.

The test was continued for four hours, commencing at 5.30 P.M., during which time all fuel and water were weighed. At the time of commencing the test a start was made with fresh fires, clean ash pits and boilers, and position of water at a known point in boilers.

Calorimeter tests were also taken throughout the time of testing.

The boilers were of the ordinary tubular kind, 5 feet diameter, 13 feet long, with domes. The fuel used was Jackson Hill slack coal, quite wet. The valves of both engines, as shown by the indicator cards, were slightly deranged, the head end of the cylinders in both engines doing the greatest proportion of the work. As one engine run 23½ hours out of the 24 each day, we had no means of taking preliminary cards in time to adjust the valves before making the test. The test was made Friday evening, December 6, 1889.

TEST OF COMPOUND CONDENSING ENGINES AT ADRIAN, MICH., WITH WATERS' THROTTLING GOVERNOR, LINKED VALVE MOTION.

NORTH ENGINE.

High-pressure cylinder, 14" diameter, 20" stroke.
Low-pressure cylinder, 21" diameter, 20" stroke.
Indicator cards from high-pressure, 40 spring.
Indicator cards from low-pressure, 20 spring.

No.	Amperes on Gauge.	Rev.	Boiler Pres.	Vacuum by Gauge. Inches.	High-Pressure.			Low-Pressure.		
					M.E.P.		I.H.P.	M.E.P.		I.H.P.
					Hd.	Cr.		Hd.	Cr.	
1	0	180	88	27	16.4	14.8	43.21	5.0	4.0	27.3
2	20	176	85	27	16.0	14.0	40.61	5.6	4.2	30.18
3	40	186	85	27	19.1	16.8	51.92	6.0	3.7	31.02
4	40	182	82	27	21.6	19.0	57.46	6.0	3.5	30.72
5	40	182	85	27	21.9	18.8	57.90			*
6	20	186	85	27	17.5	15.3	52.00			32.00
7	0	188	83	27	13.9	12.0	37.99	5.1	4.2	31.10
8	35	188	85	27	16.7	20.0	52.40	6.0	4.0	32.90
9	70	185	80	27	22.5	22.7	65.60	†	†	36.00
10	40	182	87	27	16.7	20.0	52.40	†	†	32.10
11	40	188	80	27	15.0	20.0	51.21	†	†	30.20
	Average.	......	84.6	27	......	......	50.58	......	......	30.74

* Note.—Card lost. † M.E.P. of cards not preserved.

Total I.H.P. 81.32

Variation in load, 42%.

Variation in speed, 7%.

Average admission pressure from indicator card, high-pressure, 45.8 lbs. above the atmosphere.

Average admission pressure from indicator card, low-pressure, 1.0 lbs. above the atmosphere.

Average back-pressure from cards, high-pressure, 3 lbs. above atmosphere.

Average back-pressure from cards, low-pressure, 8 lbs. below atmosphere.

Average water-rate, computed from end of stroke on card, 21.1 lbs.

SOUTH ENGINE, ADRIAN, MICH., BUILT BY LANSING IRON AND ENGINE WORKS.

Dimensions and cards as in north engine. Waters' throttling governor, link valve motion.

No.	Rev.	Steam Pres.	Vacuum by Gauge.	High-Pressure.			Low-Pressure.		
				M.E.P.		I.H.P.	M.E.P.		I.H.P.
				Hd.	Cr.		Hd.	Cr.	
1	190	81	27½	18.7	13.1	46.50	5.2	3.9	29.66
2	195	90	27½	18.6	14.0	50.48	5.8	4.3	34.46
3	194	87	27½	18.0	14.6	49.48	6.2	4.6	37.66
4	193	88	27	18.0	14.1	47.90	5.3	3.7	29.80
5	193	90	27	17.8	13.8	46.93	5.7	4.2	32.78
6	190	80	28	17.4	13.0	44.90	6.0	3.5	30.50
7	190	87	26½	17.6	13.6	46.10	6.0	4.2	33.97
8	196	84	27	17.2	13.0	45.52	5.0	3.5	25.66
9	191	85	27	17.2	12.7	44.16	5.5	3.5	29.95
10	194	84	27	17.4	13.0	45.37	5.5	3.8	30.97
11	163	85	27	26.3	10.6	46.77			34.30
Av'age.	192.6	83.7	27¼	18.57	13.23	45.83	5.74	3.92	31.80

Total I.H.P. 77.63

Variation in load, 14%.

Variation in speed, 3%—not taking No. 11 into account, as change of speed at this point was due to change of governor.

Average admission pressure from indicator cards (Figs. 125, 126 and 127), high-pressure, 43.3 lbs.

Average admission pressure from indicator cards (Figs. 125, 126 and 127), low-pressure, .50 lbs. above atmosphere.

Average back-pressure from indicator cards, high-pressure, 3.7 lbs. above atmosphere.

Average back-pressure from indicator cards, low-pressure, 8.5 lbs. below atmosphere.

Average water rate, computed from end of stroke on cards, 22.3 lbs.

The total I.H.P. of the plant would be—counting one I.H.P. of condenser pump equal to 3.6 times that of the engine, because of the high water rates of the indicator cards—

Condenser pump	8.13
Feed-pump	2.25
Both engines	158.95
Total	169.33

ADRIAN, MICH., ELECTRIC LIGHT STATION.

Two Compound Engines, Throttling Governor.

South Engine, High-Pressure, 14" Dia.,-20" Stroke.

Spring, 40. Steam by Gauge 87 lbs.

Revolutions 194.

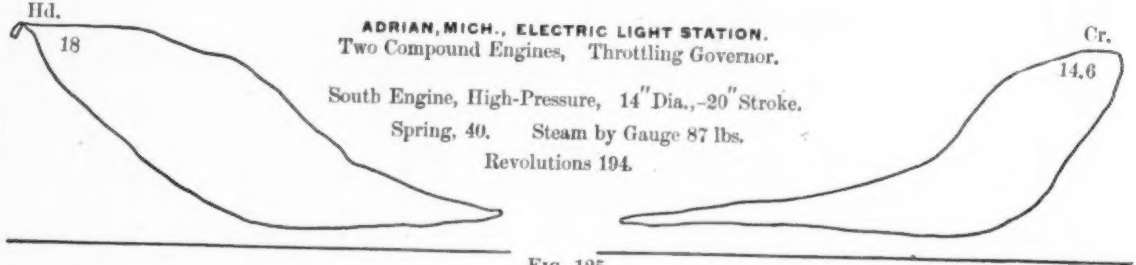
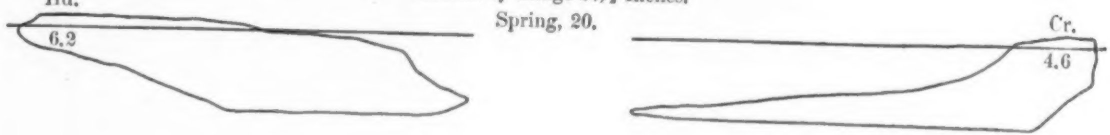


FIG. 125.

South Engine, Low-Pressure, 21" Diá.,-20" Stroke.

Vacuum by Gauge 27½ Inches.

Spring, 20.



North Engine. High Pressure, 14" Dia., 20" Stroke.

Spring 40, Steam by Gauge 83 lbs.

Revolutions 188.

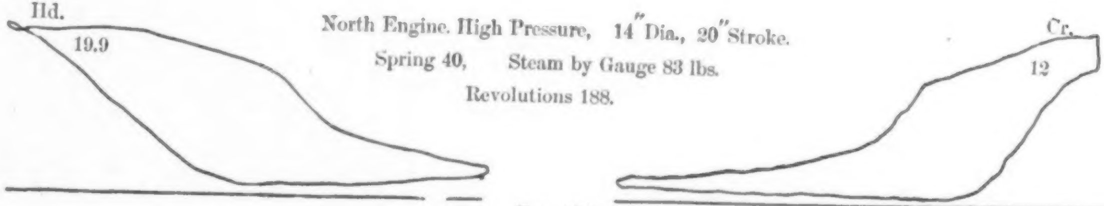


FIG. 126.

The calorimeter tests, which were conducted throughout the trial, gave results showing on the average .071% of entrained water in the steam, as follows:

CALORIMETER MEASUREMENTS.

Steam Pressure.	W Weight Condensing Water.	w Weight Wet Steam.	T Temperature Condensing Water.	T' Resultant Temperature.	H Total Heat Dry Steam.	$H-T'$	$\frac{W}{w}$	$T'-T$	l Latent Heat.	$\frac{x}{w}$ % Moisture.
84	300	5	52.2	69.9	1181.6	1112.7	60	17.7	883.4	.05
84	300	$11\frac{1}{4}$	52.2	92.2	1181.6	1089.4	26.8	40 0	883.4	.09
86	300	15	52.2	101.7	1182.1	1080.4	20	49.5	882.1	.101
86	300	20	52.2	115.7	1182.1	1066.4	15	63.5	882.1	.12
86	300	$25\frac{1}{4}$	52.2	133.5	1182.1	1048.6	11.9	81.3	882.1	.09
88	300	10	57.8	89.9	1182.5	1092.6	30	32.1	881.3	.09
88	300	15	57.8	113.5	1182.5	1069.0	20	55.7	881.3	.13
88	300	22	57.8	135.2	1182.5	1047.3	13.6	77.4	881.3	.04
80	300	15	62.3	110.4	1180.7	1070.3	20.0	48.1	885.5	.11
80	300	21	62.3	130.2	1180.7	1050.5	14.3	67.9	885.5	.09
								Average....		.071

Weight of empty barrel, 64 lbs.

WATER TEST.

The water was weighed throughout the test, the consumption being at the rate of 5,028.57 lbs. per hour; deducting from this the entrained water, the consumption of steam per hour would be 4,671.54 lbs. The consumption per I.H.P. would be 27.53 of steam per hour. During the test there was no means of pumping from the hot well, consequently water from the city water works, having a temperature of 49.6°, was pumped directly to the boiler.

FUEL CONSUMPTION.

The fuel used was wet Jackson Hill slack coal; the amount burned per hour was 811.33 lbs. At end of test the ash was weighed back, amounting to 142.66 lbs. per hour, or 17.6% of the weight; 100 lbs. of the coal was dried, and lost by evaporation 11 lbs. After making these corrections the combustible per hour was 579.33 lbs. This makes the consumption of coal 4.79 per I.H.P., and of combustible, 3.42 lbs. per I.H.P., the feed-water having a

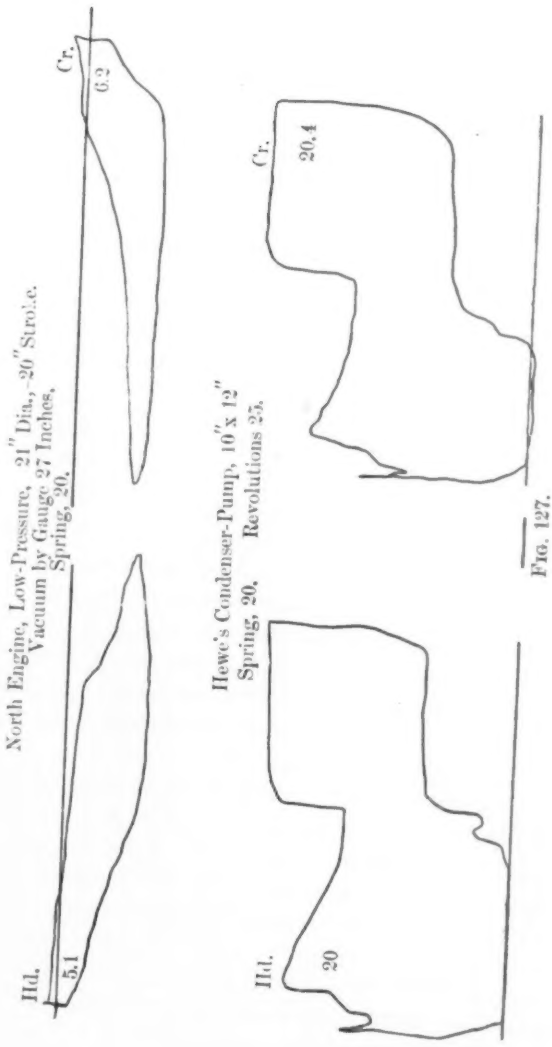

North Engine, Low-Pressure, 21" Dia., 20" Stroke.
Vacuum by Gauge 27 Inches.
Spring, 20.
Hd.
5.1
Cr.
6.2
Hewe's Condenser-Pump, 10" x 12"
Spring, 20. Revolutions 25.
Hd.
20
Cr.
20.4

FIG. 127.

temperature of 49.6°. Had the feed-water a temperature of 110°, consumption would have been as follows:

4.52 lbs. coal per I.H.P. per hour.

3.24 lbs. combustible per I.H.P. per hour.

THE BOILERS.

Ordinary tubular boilers, clean and in good condition, with no leaks, furnished with domes. The evaporation of feed-water from 49.6° was, per pound of coal, 5.75 lbs. of water; per pound of combustible, 8.05 lbs. of water. The evaporation from and at 212° would be, per pound of coal, 6.78 lbs. of water; per pound of combustible, 9.50 lbs. of water.

TEST OF TWO COMPOUND CONDENSING TRIPLE-EXPANSION ENGINES AT JACKSON ELECTRIC LIGHT STATION.

High-pressure cylinders	9½ x 20 inches.
Intermediate "	14 x 20 "
Low-pressure "	21 x 20 "

The south engine has an automatic governor attached to the valves of the 9½ and 14″ x 20″ cylinders, the valve of the low pressure cylinder being worked with a fixed eccentric.

The north engine has automatic governor attached to all three valves. The engines are so arranged that they can be worked together on the same shaft or run independently. For the purpose of testing, the engines were worked separately, but there were no possible means of obtaining separate efficiencies, as steam was supplied to both engines from same boilers.

The water rates from the indicator cards were on the average 15.92 pounds per hour for the north engine, and 15.82 pounds per hour for the south engine. The vacuum gauge of the south engine during the test showed about 24 inches, that of the north engine 21 inches; an accident which happened during the day being accountable for the poor vacuum in the north engine. Had its vacuum gauge been in usual condition, the economy of the north engine would have been better; the water rate probably would have 14.83 pounds to the I.H.P. per hour, measured from the indicator cards (Figs. 130, 131, 132).

The following are the results of the test:

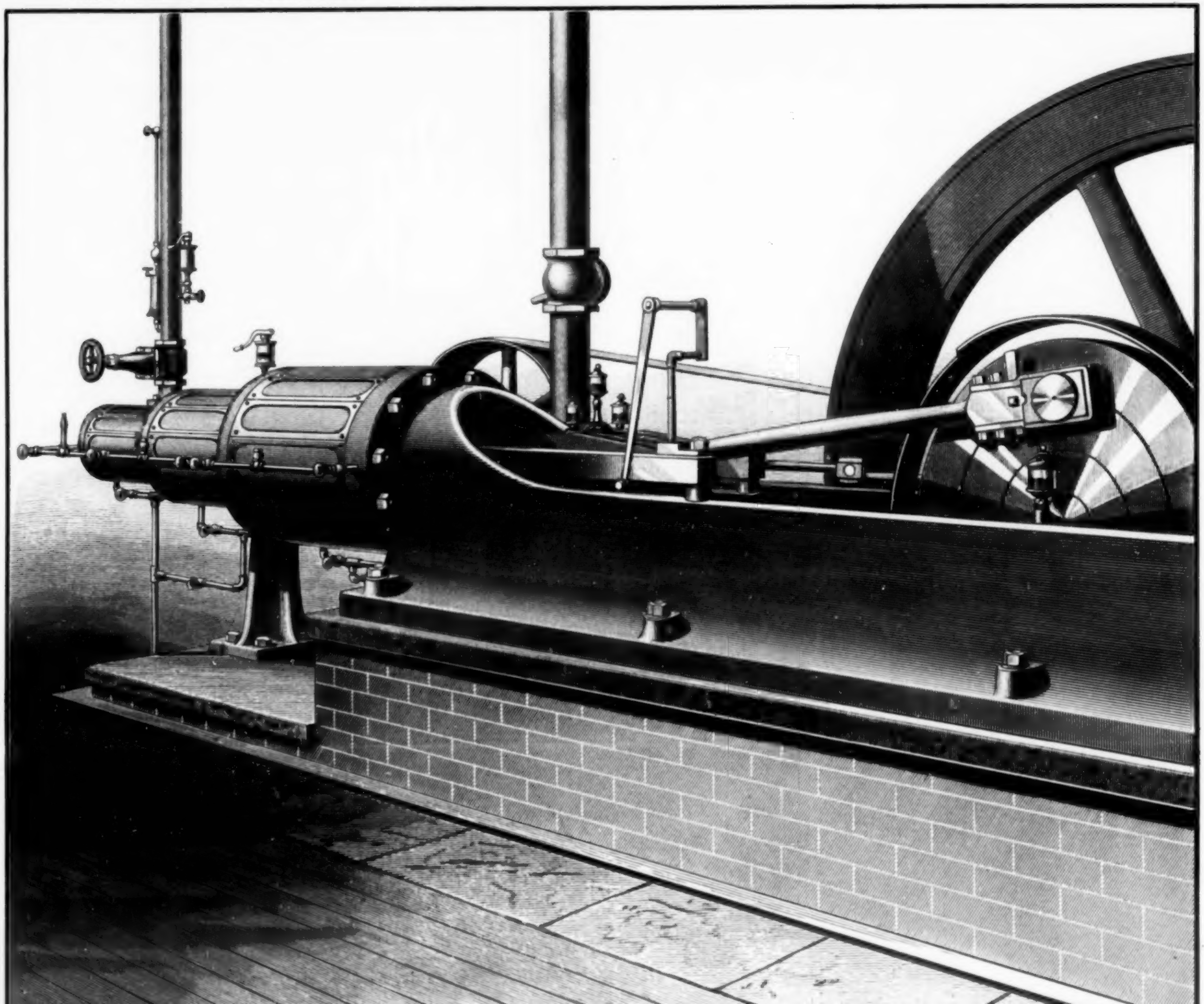


Fig. 128.

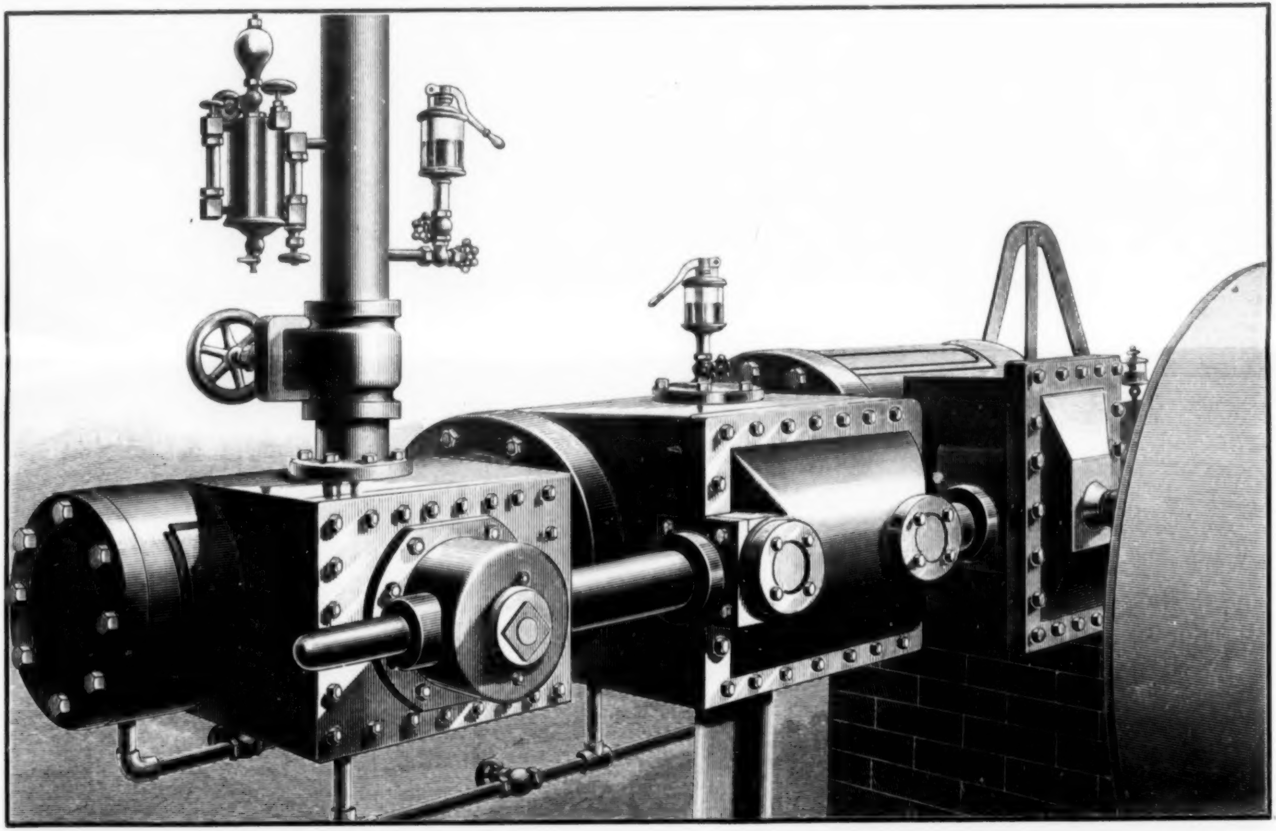


Fig. 129.

SOUTH ENGINE.

Indicator Cards.

High-pressure........................9½ x 20 inches. 60 spring.
Intermediate.........................14 x 20 " 40 "
Low-pressure.........................21 x 20 " 20 "

No.	Steam Pr.	Rev	Average M.E.P.			Indicated H.P.			Total.	Temp. Feed Water.	Vacuum by Gauge.
			High.	Int.	Low.	High.	Int.	Low.			
1	97	167	45.0	16.55	4.22	54.42	42.70	24.55	122.05	108	24
2	98	167	42.05	17.30	5.50	50.02	44.70	32.14	126.86	108	24
3	97	166	42.9	17.6	5.25	50.63	45.21	30.36	126.20	108	24
4	97	166	40.7	17.2	5.7	48.24	43.17	33.12	124.53	109	24½
5	97	166	41.3	17.5	5.6	49.07	45.16	32.51	126.74	109	24½
6	98	164	39.6	16.9	5.6	46.83	43.32	32.51	122.33	109	24
7	98	164	41.6	17.8	6.0	48.83	45.40	34.44	128.67	109	23½
8	98	164	41.4	17.8	6.1	48.59	45.40	34.56	128.55	109	23½
10	98	164	43.4	16.9	5.7	50.71	43.32	32.72	126.75	109	23½
11	98	166	41.7	17.5	6.35	49.55	45.16	36.53	131.24	110	23½
12	98	166	40.9	——	5.4	48.59		31.58		110	23½
13	98	166	39.2	16.8	5.2	46.59	43.41	30.17	120.11	110	23½
14	97	166	——	17.2	5.4		44.41	31.58		110	23½
15	98	168	39.3	16.0	5.1	47.30	41.30	35.30	123.90	110	23½
16	98	168	39.0	16.9	5.7	46.90	44.15	33.79	128.84	110	23½
Average ...			41.3	16.9	5.43	49.09	44.06	32.40	125.50		

Average steam consumption from card, 15.82 lbs. per I.H.P.
Variation in load, .09%.
Variation in speed, .024%.

NORTH ENGINE.

Indicator Cards.

High-pressure cylinder...............9½ x 20 inches. 60 spring.
Intermediate " ....14 x 20 " 40 "
Low-pressure " ....21 x 20 " 20 "

No.	Steam Pr.	Rev.	Average M.E.P.			Indicated H.P.			Total.	Vacuum by Gauge.
			High.	Int.	Low.	High.	Int.	Low.		
1	98	166	45.9	16.7	7.8	54.28	43.10	45.32	143.6	21
2	97	166	48.4	17.3	7.7	57.52	44.67	44.73	146.92	20½
3	98	166	49.4	16.65	7.1	58.70	42.98	41.25	142.93	21
4	98	166	44.1	16.9	7.4	52.40	43.53	43.0	138.93	21
5	97	167	44.2	17.5	7.25	52.90	45.43	42.65	140.98	21
6	97	167	45.9	16.9	6.8	54.61	43.81	39.76	138.18	21
7	98	167	46.0	16.4	7.7	55.00	42.55	45.01	142.56	20¾
12	98	165	43.9	16.3	7.2	52.83	42.57	41.58	136.97	21
13	98	162	46.3	16.5	6.9	56.68	42.04	39.64	138.46	21
14	97	163	46.3	16.0	7.6	56.68	40.80	44.40	141.88	21
16	97	167	45.45	15.9	7.0	54.34	41.24	40.92	136.50	21
17	98	167	45.4	16.3	7.8	54.24	42.30	45.59	142.10	21
Average			45.9	16.6	7.35	55.20	42.92	42.81	140.93	

Average steam consumption from card, 15.92 pounds per I.H.P.
Variation in load, .079%.
Variation in speed, .031%.

CALORIMETER TEST.

$$\text{Percentage moisture} = \frac{x}{w} = \frac{H - T' - \frac{W}{w}(T' - T)}{l}$$

Steam Pressure.	W Weight Condensing Water.	w Weight Wet Steam.	T Temperature Condensing Water.	T' Temperature of Result.	H Total Heat in Dry Steam.	$H-T'$	$\frac{W}{w}$	$T'-T$	Per Cent. of Moisture.	l
77	300	15	57.4	107.8	1180.0	1072.2	20	50.4	.07	887
84	300	15	62.2	115.5	1181.6	1066.1	20	53.3	.00	833.9
98	300	15	56.8	105.6	1189.6	1079.0	20	49.8	.09	876.2
97	300	15	53.8	106.4	1184.5	1078.1	20	52.8	.07	877
97	300	15	54.6	109.2	1184.5	1075.3	20	54.6	.02	877
97½	300	15	53.0	104.2	1184.5	1080.3	20	51.2	.06	877
97½	300	10	53.0	88.8	1184.5	1095.7	30	35.8	.025	877
97½	300	15	53	105.5	1184.5	1079.0	20	52.5	.03	877
97½	300	25	53	137.0	1184.5	1047.5	12	84.0	.04	877
97½	300	30	53	153.0	1184.5	1031.5	10	100	.035	877

Average steam, .034% wet.

WATER CONSUMPTION.

The amount of water used by the engines per hour equalled 4,960 lbs.; this is equal to 18.61 lbs. per I.H.P. per hour. Deducting entrained water, .034%, the water consumption of the engine was 17.98 lbs. per I.H.P.

EVAPORATION OF BOILERS.

74.19 lbs. of water raised from a temperature of 110° and converted into steam at a pressure of 97 lbs. for each gallon of oil burned. This would make the evaporation work of one gallon of oil 82.49 lbs. of water.

If the weight of a gallon of oil is 7.4 lbs., the evaporation would be 11.15 lbs. of water to one pound of combustible.

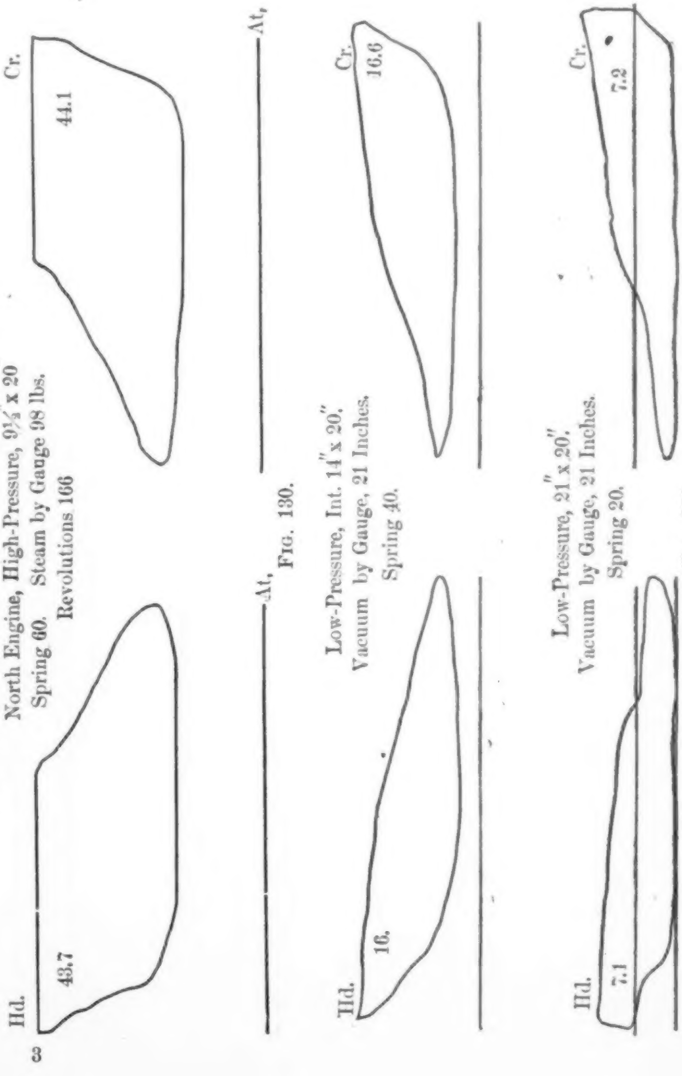
FUEL PER ONE HORSE-POWER.

The consumption of oil was 80.72 gallons per hour for 266.43 I.H.P.; this is equal to .303 of one gallon per I.H.P. per hour. An approximation of the fuel used per I.H.P., extending over a long time, by Mr. James Foote, was as follows:

WORK OF STATION.

275 H.P.	for 5	hours	= 1,375 H.P.	for one hour.
230 H.P.	" $8\frac{1}{2}$	"	= 1,955 H.P.	" " "
30 H.P.	" 3	"	= 90 H.P.	" " "

This would give, on the average, 3.37 H.P. from one gallon of oil, or an amount very nearly the same as obtained in the test.



JACKSON, MICH., ELECTRIC LIGHT STATION.
Two Compound Engines,—Automatic, 9½"-14"-21" x 20".
North Engine, High-Pressure, 9½" x 20"
Spring 60. Steam by Gauge 98 lbs.
Revolutions 166

FIG. 130.

Low-Pressure, Int. 14" x 20".
Vacuum by Gauge, 21 Inches.
Spring 40.

Low-Pressure, 21" x 20".
Vacuum by Gauge, 21 Inches.
Spring 20.

FIG. 131.

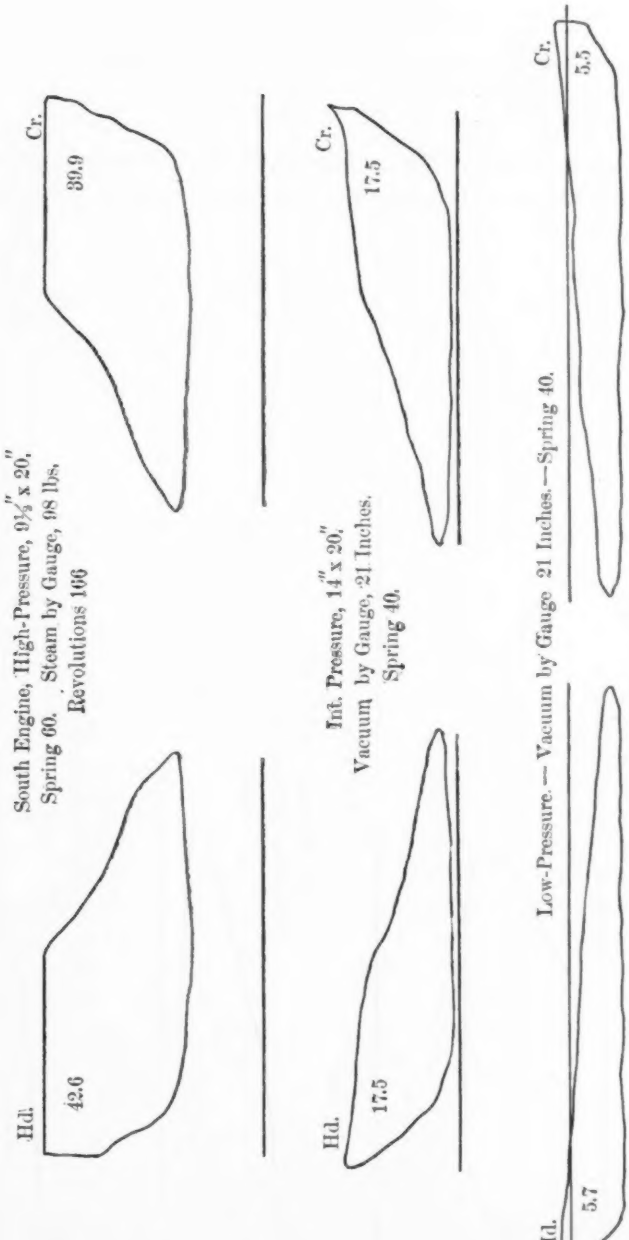

South Engine, High-Pressure, 9½" x 20".
Spring 60. Steam by Gauge, 98 lbs.
Revolutions 166
Hd.
42.6
Cr.
39.9
Int. Pressure, 14" x 20".
Vacuum by Gauge, 21 Inches.
Spring 40.
Hd.
17.5
Cr.
17.5
Low-Pressure.— Vacuum by Gauge 21 Inches.—Spring 40.
Hd.
5.7
Cr.
5.5

FIG. 132.

JACKSON ELECTRIC LIGHT STATION TRIPLE EXPANSION ENGINES.

AVERAGE MEASUREMENTS FROM INDICATOR CARDS. (Figs. 133, 134.)

SOUTH ENGINE.

	Average Steam Pressure.	Average Vacuum by Gauge.	High-Pressure.	Inter-medi-ate.	Low-Pres-sure.
Admission pressure above atmosphere	97½ lbs.	24 in.	94.2	31.2	0.4
Release pressure	....	....	45.5	6.6	5.16
Back-pressure above atmosphere	....	....	31.2	0.4	8.1
Absolute back-pressure shown by gauge	....	....	....	......	2.7
Absolute back-pressure in cylinders	....	....	45.9	15.1	5.6
M.E.P	....	....	41.39	16.9	5.43
NORTH ENGINE.					
Admission pressure above atmosphere	97½ lbs.	21 in.	94.0	33.8	5.75
Release pressure	....	....	53.6	12.3	−2.84
Back-pressure above atmosphere	....	....	33.0	5.4	−5.6
Absolute back-pressure shown by gauge	....	....	....	......	4.2
Absolute back-pressure in cylinders	....	....	47.7	21.1	3.5
M.E.P	....	....	45.9	16.6	7.35

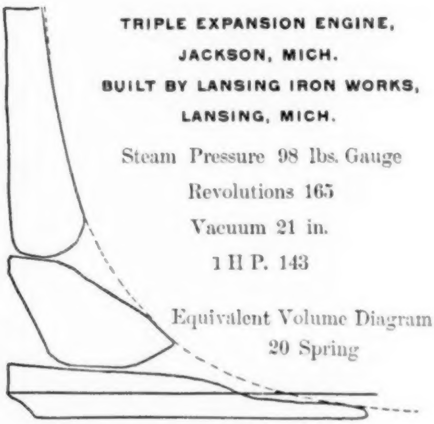


FIG. 133.

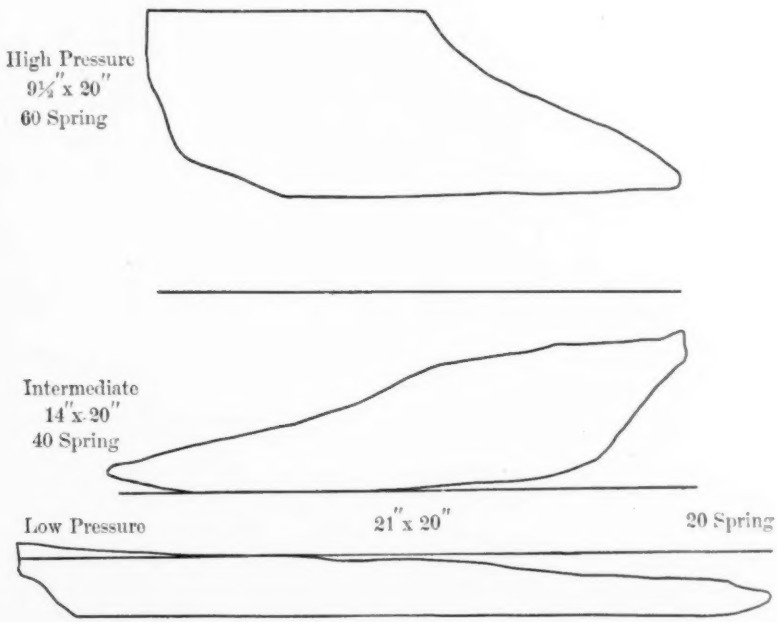


FIG. 134.

SUMMARY OF RESULTS.

	Simple Automatic.	Corliss Condensing.	Compound Non-Condensing.		Compound Condensing. Throttling Governor, Independent Condenser.		Compound Condensing. Automatic Governor Belted Condenser Pump.			Triple Expansion.	
Location	Jonesville, Mich.	Lansing, Mich.	Lansing, Mich.		Adrian, Mich.		Albion, Mich.	Lansing, Mich.		Jackson, Mich.	
Dimensions—High / Low	14 × 20	14 × 36	14 × 20 21 × 20		14 × 20 21 × 20		9¼ × 20 21 × 20	14 × 20 21 × 20		9¼ × 20 14 × 20 21 × 20	
Number of Engines	1	1	1		2		1	2		2	
Steam Pressure	70.3	69.	80.		83.7		130.5	100.1		98.0	
			1st Test.	2d Test.							
Steam per I.H.P.	30.3	26.87	32.8	32.6	27.53		18.77	16.57		17.98	
Per cent. Entrained Water	....	0.67	.05	.05	.071		5 6	.11		.034	
Coal per I.H.P.	5.34	4.63	5.2	5.12	4.52		3.05	1.96		Oil. .303 gal.	
Combustible per I.H.P.	4.76	3.61	4.2	4.12	3.24		1.97	1.69			
Evaporation per lb. Coal	6.70	6.60	7.4	7.46	6.78		6.90	9.06			
Evapor'n per lb. Combustible	7.49	8.27	9.2	9.56	9.50		10.66	10.45		11.15	
					No. Eng.	So. Eng.		East Eng.	West Eng.	No. Eng.	So. Eng.
Vacuum in inches by Gauge	....	27	....		27.2		26.4	23.0	23.1	21.0	24.0
I.H.P. High					50.58	45.83	56.04	68.28	43.30	55.2 42.9	49.0 44.1
I.H.P. Low					30.74	31.80	39.43	69.04	47.50	46.8	22.4
								137.82	90.8	140.9	125.5
I.H.P. Total	45.45	61.99	69.9	68.75	81.32	77.62	95.47	227.4		266.4	
Revolutions	171.5	95	149.8		184.1	193	160.5	172	172	165.7	165.6
Kind of Work	Flour Mill.	Flour Mill	Wheel Fact.		Electric Light Engines.						
Measurements from Indicator Cards.											
Steam per I.H.P.	24.5	22.	25.8		221	221.3	15.84	15.8	19.4	15.9	15.82
High-Pressure Cards. Admission	63.3	58.	6.8		45.8	43.3	113.3	84	70	94	94.2
High-Pressure Cards. Release	8.2	46.	20.		9.6	8.5	30.0	22	19	53.6	45.5
High-Pressure Cards. Back-Pressure	1.0	11.	15.		3.0	3.7	11.0	14	12.5	33.0	31.2
Low-Pressure Cards. Admission			15.		1.0	¼	7.3	10.1	6.0	39.8 12.3 5.4	31.6 6.6 0.4
Low-Pressure Cards. Release			4.		−6.0	−5	−5.45	−3.7	−5.0	5.75 −2.84	0.4 −5.16
Low-Pressure Cards. Back-Pressure			2.5		−8.0	−8.5	−8.00	−7.5	−8.0	−5.6	−8.10
Ratio of Cylinders			1 to 2¼		1 to 2¼		1 to 4.88	1 to 2.25		1 to 2.25 2.25 to 4.88	
Total Expansions	5	15	22.5		13.5	18.0	36	26.1 to 56		20.5	31.3
Fuel—kind	Jackson Hill Lump.	Soft Nut.	Soft Lump.		Jackson Hill Slack.		Soft Slack.	Soft Lump and Coke.		Crude Oil.	
Per cent. Waste	.11	.22	.20		.286		.358	.133			

DISCUSSION.

Prof. D. S. Jacobus.—I would like to ask about the percentage of entrained water. The author has found about $6\frac{7}{10}\%$ in one case, and some of the tests appear to show a still greater amount. This is higher than one will ordinarily find with the engine standing near the boiler. The paper states that the pipe is covered, and under this set of conditions I should think that was a very high percentage of priming.

Mr. Carpenter.—The calorimeter tests are described in the

paper. In that engine the boiler was about seventy-five feet distant. I think the calorimeter tests ran quite evenly. I had a very good pair of scales and a standard thermometer 24 inches long, made by Henry J. Green. Nearly all the engines showed a great deal of entrained water, although I think with this exception all the pipes were uncovered.

Prof. Jacobus.—Were the results obtained with the barrel calorimeter checked in any way, such as by observing the appearance of the steam on issuing from an orifice? I have made a number of tests with barrel calorimeters, and I find, by altering the conditions, under certain circumstances widely different results may be obtained. Experiments made in order to determine the calorific effect of a wooden barrel do not, as a rule, agree very well with each other.

Mr. Carpenter.—I made these calorimeter tests as carefully as I could. I know that difficulty very well with the barrel calorimeter, and I have also experienced the same irregularities with other calorimeters. I had no way of checking the results, but believe them to be quite accurate. I made no correction for temperature-equivalent of the barrel, as it would in any event have scarcely changed the results. I have no question in my mind but that all calorimeter measurements are likely to be somewhat irregular. I have often noticed this. I presume all have felt it. I have never been satisfied in my own mind whether that was the fault of the calorimeter or the flowing over of entrained water. I have recently concluded that it was very likely to be the actual flowing over of the water; because I know, if the water-line changes a little or the action of the engine requires a little more steam, the effect is to make a pulsation or wave-motion which, I think, might draw over the water. The calorimeter measurements were made in that way, and as carefully as possible under the circumstances. I believe the results in that especial case to be correct.

Mr. T. J. Borden.—Will Mr. Carpenter state to us how that pipe was taken from the boiler; whether it was taken from the shell without any enlargement of the opening?

Mr. Carpenter.—This particular pipe was taken directly from the boiler, carried up to a height of three, possibly four, feet, and then carried over to the engine, which was in an adjoining room; and there were three or four elbows—I am not certain about the number—making a total distance of about seventy-five feet.

It was covered with a composition invented by the miller, and made by simply putting on flour, bran, and middlings mixed with water, making an adhesive coat which was almost as firm as a plastered wall, and it had been on several years.

Mr. Nagle.—I would suggest to Mr. Carpenter that, when he uses the calorimeter, he pursue the method Mr. Barrus does in the use of his calorimeter—that he try it also when the engine is not using steam; and that would be a check on all his calculations.

Mr. Carpenter.—It would not check it at all. The wet steam in the case under discussion was no doubt due to the high waterline carried by the firemen, in spite of all my remonstrances.

Mr. Nagle.—Have you ever tried it? Do you get dry steam?

Mr. Carpenter.—I have, and sometimes get dry steam.

Mr. Webber.—I would suggest that Mr. Carpenter do not use a barrel calorimeter at all. If he used a Barrus or Peabody calorimeter he would get better results.

Mr. Carpenter.—These, especially the old forms, would require drier steam than we get. I believe the barrel calorimeter, carefully handled, will give fairly accurate results.

CCLXXXV.

PÉCLET'S TREATMENT OF CHIMNEY DRAUGHT.

BY J. BURKITT WEBB, HOBOKEN, N. J.

(Member of the Society.)

In the discussion upon chimney draught at the last meeting of this Society I commented briefly upon the treatment in Péclet and promised a paper upon the subject at this meeting. My part in that discussion should, therefore, be read in connection with what follows here, to avoid needless repetition.

The portion of that treatment which it was the intention to discuss may be translated freely as follows:*

"538. If the pressure be expressed in a column of air at θ degrees, we have, x being the height of the column,

$$x = \frac{Ha\,(t - \theta)}{1 + at};$$

if it be expressed in a column at t degrees, it will be

$$\frac{Ha\,(t - \theta)}{1 + a\theta}.$$

. . . .

Péclet, in his last edition, chose the first in accordance with some experiments, not very complete, and which seem insufficient to decide the question.

"539. The mechanical theory of heat allows the difficulty to be solved.

Suppose that at the base of the chimney the atmospheric pressure expressed in kilogrammes per square meter is p_0 and that θ is the temperature of the outside air, also that p is the pressure at the top of the chimney and t the temperature of the air which flows out.

If we consider an infinitely thin slice of air in the chimney, dh

* "*Traité de la Chaleur* par E. Péclet, Quatrième édition, publiée par A. Hudelo." Vol. I., page 279.

will be its height and dt the variation of the temperature of the air in its passage across the slice; supposing no heat to be lost through the walls of the chimney, and putting u for the velocity, the increase of external kinetic energy of one kilogramme of air passing through one square meter of the section of the chimney will be $udu \div g$, the work equivalent of the heat which disappears will be $dQ \div A$, and the work done against gravity will be dh. The equation for the energy will then be

$$\frac{udu}{g} = \frac{dQ}{A} - dh. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

On the other hand, if we consider the air entering the chimney at the temperature θ and at the pressure p_o, it must be heated by the fire, under the constant pressure p_o, to a temperature t', such that after expanding adiabatically it will reach the top of the chimney with the temperature t and pressure p, so that we shall have

$$\frac{1 + at}{1 + at'} = \left(\frac{p}{p_o}\right)^{\frac{k-1}{k}}$$

or, if we put $\dfrac{k-1}{k} = n$

$$\frac{1 + at'}{1 + at} = \frac{p_o^n}{p^n}$$

and finally

$$t' - t = \frac{1 + at}{a} \cdot \frac{p_o^n - p^n}{p^n}.$$

The equation (1) being integrated, gives for the velocity of exit of the air

$$\frac{u^2}{2g} = \int_t^{t'} \frac{dQ}{A} - H. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

H being the height of the chimney, whence

$$\frac{u^2}{2g} = \frac{C}{A}(t' - t) - H, . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and consequently

$$\frac{u^2}{2g} = \frac{C}{A} \cdot \frac{1 + at}{a} \cdot \frac{p_o^n - p^n}{p^n} - H.$$

If D is the mean weight, in kilogrammes per cubic meter, of a column of the exterior air the height of the chimney, we shall have

$$p_o = p + HD;$$

and, if we put

$$p = m\ HD,$$

we shall have

$$p_o = (1 + m)\ HD = m\left(1 + \frac{1}{m}\right) HD,$$

whence

$$p_o^n = m^n\left(1 + n\frac{1}{m} + n(n-1)\frac{1}{m^2} + \ldots\right) H^n D^n,$$

and, as m is a very large number, we shall have

$$p_o^n = m^n H^n D^n\left(1 + \frac{n}{m}\right),$$

whence

$$\frac{p_o^n - p^n}{p^n} = \frac{n}{m}.$$

Substituting this in the value of u it becomes

$$\frac{u^2}{2g} = \frac{Cn}{A}\cdot\frac{1 + at}{am} - H; \quad \ldots \ldots \quad (4)$$

but we have

$$m = \frac{p}{HD}\cdot\frac{Cn}{A}\cdot = \frac{0.76 \times 13.6a}{1.293};$$

and if we suppose

$$D = \frac{1.293\ (p + p_o)}{2\ (1 + a\theta)\ 0.76 \times 13.6},$$

we have, by substituting these values,

$$\frac{u^2}{2g} = \frac{p + p_o}{2p} H \frac{1 + at}{1 + a\theta} - H.$$

Now, without sensible error, we may put

$$\frac{p + p_o}{2p} = 1,$$

whence $$\frac{u^2}{2g} = \frac{Ha\ (t - \theta)}{1 + a\theta} \quad \ldots \ldots \ldots \quad (5)$$

The last formula is, therefore, the one that is to be used.

"540. Accepting this formula there results for the velocity of the cold air

$$v = \frac{1 + a\theta}{1 + at}\sqrt{\frac{2gH(t - \theta)}{1 + a\theta}}$$

or,

$$v = \frac{0.268}{1 + at}\sqrt{H(t - \theta)(1 + a\theta)}, \quad . \quad . \quad . \quad (A)$$

the velocity of the hot air being

$$v' = 0.268\sqrt{\frac{H(t - \theta)}{1 + a\theta}}. \quad . \quad . \quad . \quad . \quad (B)$$

"544. If we compare the results deduced from formula (A) with those obtained by employing the formula, which supposes the head expressed in cold air, we have, for $\theta =$ zero.

$$v = \sqrt{2gH}\sqrt{\frac{at}{1 + at}},$$

and if we make t successively equal to 50°, 100°, 150°, 200°, 250°, 300°, 350°, 400°, 500°, 1000°, 1500°, 2000°, we find for the values of the second factors .39, .51, .57, .64, .68, .71, .74, .76, .80, .88, .91, .93.

As far as 100° the results deduced from the two formulas differ but little, and for a temperature of 300°, which is rarely exceeded in practice, the values are in the ratio of 7 to 5."

It is difficult to see why certain parts of this analysis may not be shortened to advantage; we shall therefore venture to remodel it and shall be able to examine it more satisfactorily in the reduced form. The equations will be numbered to correspond with the previous ones, and centigrade degrees will be adhered to.

In the paragraph following equation (1) it will be better to introduce τ_0, τ and τ' for θ, t and t', and then to proceed immediately with the integration of (1), thus:

Integrating between the limits τ and τ' we have

$$\frac{u^2}{2g} = \int_{\tau}^{\tau'} \frac{dQ}{A} - H, \quad . \quad . \quad . \quad . \quad . \quad . \quad (2')$$

or,

$$\frac{u^2}{2g} = \frac{C}{A}(\tau' - \tau) - H, \quad . \quad . \quad . \quad (3')$$

where H is the height of the chimney.

By the law of adiabatic expansion

$$\frac{\tau'}{\tau} = \frac{p_0^{\,n}}{p^n},$$

from which, by division,

$$\tau' - \tau = \tau\frac{p_0^{\,n} - p^n}{p^n} = \tau\left[\left(\frac{p_0}{p}\right)^n - 1\right].$$

Here $n = \frac{k-1}{k}$, and if we put $p_0 = p + \varDelta p$ we shall get

$$\tau' - \tau = \tau\left[\left(1 + \frac{\varDelta p}{p}\right)^n - 1\right]$$

$$= \tau\left[\left(1 + n\frac{\varDelta p}{p} + \text{etc.}\right) - 1\right] = \tau\, n\,\frac{\varDelta p}{p}$$

$$= \tau n \frac{\varDelta p}{p}$$

approximately, the higher powers of $\frac{\varDelta p}{p}$ being omitted from the development as insignificant.

Substituting this value of $\tau' - \tau$ in (3') there results

$$\frac{u^2}{2g} = \frac{Cn\tau}{Ap}\varDelta p - H, \quad . \quad . \quad . \quad . \quad . \quad . \quad (4')$$

which is in terms of the temperature and pressure of the air issuing from the top of the chimney.

$\frac{C}{A}$ is the work equivalent of the heat required to raise a kilogramme of air 1°, and $Cn \div A$ is the external work, or work of dilatation per degree, $Cn\tau \div A$ is then the work of dilatation for τ degrees, which is equal to pv; therefore $Cn\tau \div Ap = v = \frac{1}{\delta}$, v being the volume of a kilogramme of hot air at the top of the chimney and δ the weight of a cubic meter of the same.

We can now write (4') in the form

$$\frac{u^2}{2g} = \frac{\varDelta p}{\delta} - H = H' - H, \quad . \quad . \quad . \quad . \quad . \quad (4\tfrac{1}{2})$$

Now $\varDelta p$ is the weight of a column of cold air whose section is unity, height H, and temperature θ; therefore $\varDelta p \div \delta$ is the height H' of a column of hot air of the same weight and section.

and whose temperature is t. The velocity u is thus seen to be due to the excess in height of the column of hot air over the cold-air column. The expression may be further simplified by expressing H' in terms of H and the temperatures,

$$H' = H\frac{\tau}{\tau_0},$$

and this reduces the last equation to

$$\frac{u^2}{2g} = \frac{\tau - \tau_0}{\tau}H = \frac{Ha\,(t-\theta)}{1 + a\theta} \quad . \quad . \quad . \quad . \quad (5')$$

Comparing the analysis with this modified form we see, first, that the introduction of m is no simplification and is unnecessary; second, that the introduction of the mean density D of the outside air is not only useless but misleading. The formula for D is only approximate at best, and when $(p + p_0) \div 2p$ is put equal to 1 ($i.e.$, $p = p_0$) it changes D to the density of the cold air at the top of the chimney, so that the plainest way would have been to take the density there in the first place as sufficiently exact, instead of leading up to it by two approximations, the latter of which ($p_0 = p$) would be inadmissible in other parts of the analysis. But D is not wanted, because $\Delta p = p_0 - p$ is evidently a more available and exact value for the weight of the cold-air column than HD is, and because the densities disappear from the final equations.

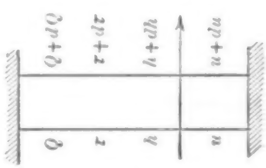


FIG. 135.

Thus much in reference to the algebraic features of the analysis; we shall now examine the thermo-dynamic part.

Fig. 135 is drawn to illustrate the suppositions of the paragraph preceding equation (1). Of course dh is plus, and, supposing du plus, dQ will be plus and $d\tau$ minus. Q is the quantity of heat furnished by the air in cooling $d\tau$, and this is the only change of heat involved in crossing the differential slice. But equation (1) is not complete, an important term being omitted; we shall therefore produce the correct equation and compare it with (1).

The only way in which a velocity can be increased or a weight lifted is by the application of a force in the proper direction. The force in this case must be an excess of pressure at the lower face of

the slice over that at the upper, and the only way in which the quantity dQ of heat is concerned is in assisting to create this difference of pressure; p being the pressure at the lower surface of the slice, that at the upper is $p + dp$, dp being minus. The work done on a kilogramme of air passing across the slice is therefore the product of $-dp$ by the volume of one kilogramme or $-vdp$, so that we have

$$\frac{u^2}{2g} + dh = -\,vdp \quad . \quad . \quad . \quad . \quad . \quad . \quad (1'')$$

the minus sign being needed to reverse the intrinsic sign of dp and make the expression for the work plus.

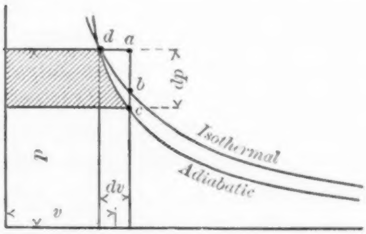


FIG. 136.

The shaded area in Fig. 136 shows this work corresponding to the expansion of the air along the adiabatic curve from d to c. Now this adiabatic expansion is a compound of the isothermal expansion db and the fall of temperature bc at constant volume; in the same way the fall of pressure dp is equal to the isothermal fall ab plus the further fall bc to the adiabatic, between which there exists the relation

$$bc : ac = 1 : \gamma = C_1 : C.$$

The quantity of heat dQ corresponds to the fall of pressure bc, being equal to $v.\,bc = C_1 d\tau$, comparing which with the term in (1''), whose value is

$$-\,vdp = C\,d\tau,$$

it appears that, to make (1) true, its right hand member should have added to it the term

$$(C - C_1)\,d\tau.$$

The heat represented by this expression is part of the latent heat given to the air in heating it from τ_0 to τ'—*i.e.*, from θ to t'—at which time it was converted into work and stored up by gravity. While the addition of this quantity would make (1) correct, it could only enter as representing a quantity of work, not heat, and, indeed, this is the only way in which dQ can enter, both being included in the work $-\,vdp$.

Equation (3) is the integral equation of (1″) and is not, therefore, the integral of (1′); so that the analysis contains not only what we cannot help regarding as an error, but a second compensating one.

It is to be noticed that in similar equations (page 169) the term has not been omitted.

The meaning of the artificial temperature t', or τ', which is used as a limit in equation (2), must now be considered. It does not appear from the analysis that the author had its meaning clearly in mind.

It will be seen that this temperature has no counterpart in an actual chimney and that its introduction requires a considerable modification of the problem. The use of t', therefore, with no proof that such a modification is admissible, affects the reliability of the

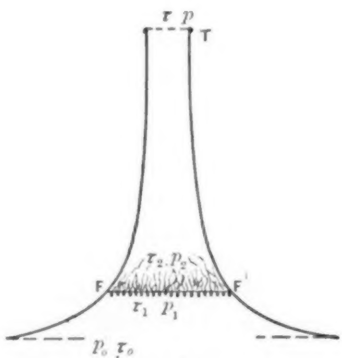


FIG. 137.

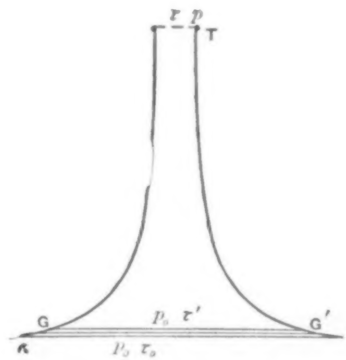


FIG. 138.

argument and might raise doubts as to the accuracy of the result, were it not otherwise known to be correct.

Fig. 137 corresponds to an actual chimney in which the air acquires a velocity before it reaches the fire FF', and in doing so experiences an adiabatic fall of temperature and pressure from τ_0 to τ_1 and from p_0 to p_1. The fire then heats it to τ_2 and greatly increases its volume, so that the velocity must correspondingly increase, which requires the further fall of pressure from p_1 to p. The pressure will then fall still further to p, producing a further increase of velocity, if the cross-section decreases before reaching the top of the chimney. If the air is supplied freely beneath the grate the most of the velocity is produced by the fall of pressure $p_1 - p$. It should be borne in mind that the only

thing which can produce velocity is force—*i. e.*, here, the difference of pressure; heat or the difference of temperature cannot do it except by producing a difference of pressure. Friction is also supposed to be nothing, so that all the difference of pressure is available to produce velocity.

Now τ_2 is not the temperature τ', and to understand the latter purely artificial temperature we must have recourse to Fig. 138. Here the fire is supposed to be removed to GG' or, indeed, removed altogether and in its place a grating of wire substituted, which can be heated electrically to any desired temperature. The cross-section GG' is supposed infinite so that the wires may offer no obstruction to the passage of the air. There will then be no finite change but that of temperature and volume as the air passes the wires; the pressure on both sides will be p_o, and the velocity zero. τ' is then the temperature to which the air must be heated in order that it may cool off to τ by the time it reaches the chimney top, and τ' does not agree with any particular temperature in an actual chimney.

This view of the problem, admitting its correctness, is a very interesting one, for it reduces the question to that of the flow of the air from a reservoir where the pressure is p_o and temperature t'; in fact, the conditions between G and T are precisely the same as in that case. The only difference between the two is in the way in which the air at p_o and temperature t' is supplied. In the reservoir there is a quantity of such air ready stored up, by which the flow is sustained, while in the present problem the air is put into the condition p_o t', as fast as it is needed, by passing the heated grate.

It would seem that, with this view clearly in mind, the author might have avoided the duplication of the analysis by referring to a previous treatment of the subject. Compare the analysis with that on pages 168, *et. seq.*

In a chimney 100 metres high, if τ were double τ_o, $\tau' - \tau$ would be about two degrees centigrade.

Equation ($4\frac{1}{2}$) not only shows that the thermo-dynamic part of the analysis has been eliminated, but indicates that the true origin of the velocity is the head $H' - H$, which is equivalent to saying that the better way to treat the question is to start with the general equation

$$u = \sqrt{2gh}$$

and substitute the proper value of h, which is not at all ambiguous.

(See Paper No. CCCLXXXVI., *The Mechanical Theory of Chimney Draught.*)

With so small a value as $\tau' - \tau < 2°$ it must be evident that, in finding h by the ordinary method, it is sufficiently accurate to suppose all the air in the chimney to be at the temperature τ, thus taking no account of its slight change in density between the top and bottom.

[NOTE.—*This paper received discussion jointly with the other paper by the same author, entitled "The Mechanical Theory of Chimney Draught," printed as No. CCCLXXXVI. of the papers of this meeting.*]

CCCLXXXVI.

THE MECHANICAL THEORY OF CHIMNEY DRAUGHT.

BY J. BURKITT WEBB, HOBOKEN, N. J.
(Member of the Society.)

In a previous paper the statement was made that when the draught of a chimney is produced by the heat of a furnace its velocity does not depend directly upon the expenditure of heat, and therefore is not a problem in thermo-dynamics. It was further stated that the velocity depended directly upon the action of gravity, and the phenomenon was likened to that of a clock, whose speed of running is controlled by a pendulum. By the winding up of a weight energy is stored and thus placed at the disposal of the mechanism, but the rate at which the mechanism uses the energy is controlled by gravity.

The chimney may also be compared to a vessel kept full of water, which is allowed to flow out of a hole at the bottom; the velocity of the issuing jet depends upon the head of water in the vessel, and so in the chimney the velocity of the issuing hot air depends upon a head of hot air. In both cases, if the flow is to be kept up continuously, as much must be supplied as flows out, and this requires water to be lifted in the one case and air in the other.

We shall in this paper show exactly how gravity acts in producing the velocity of the hot air in a chimney, and how the heat acts to keep gravity wound up, so to speak.

Let *CD* (Fig. 139) be a chimney, say, 330 feet high by 1 foot-square section, full of hot air, and let *AB* be a shaft 100 feet square, of the same height, full of cold air, *BC* being a passage connecting the two. For the sake of simplicity we shall assume that the air in the chimney is so hot that its density is but half that of the cold air, and that after the air is heated no heat is abstracted by a boiler or lost through the walls. By placing the grate near the bottom of the large cold-air shaft it will be so large as to offer no perceptible obstruction to the passage of the air, and we

may also suppose that the grate bars are simply wires capable of being heated by electricity, so that there will be no coals to obstruct the flow and so that the heat can be turned on and off by making and breaking the circuit. Let EF be such a grating of wires.

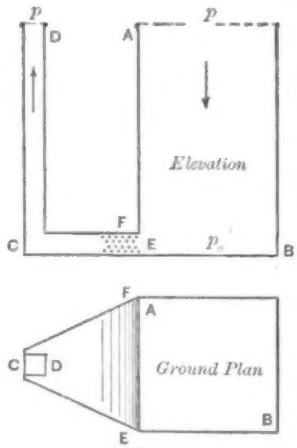


Fig. 139.

The cold-air shaft having 10,000 times the section of the chimney, and the cold air only half the density of the hot, the velocity in the shaft will be but $\frac{1}{20000}$ of that in the chimney, and therefore neglectable.

It must be evident at once that, if the chimney be in full draught and if the current be broken so as to stop the supply of heat, no diminution of the velocity will result until all the hot air between FE and C has passed into the chimney. But this space FEC can be made of any capacity, so that the effect upon the velocity of stopping the supply of heat can be postponed indefinitely, which proves that *the heating of the air has no direct effect in producing the velocity.*

We have, therefore, to consider separately

The effect of heating the air and the production of the velocity.

The pressure of the atmosphere at A and D being the same may be left out of the problem.

THE EFFECT OF HEATING THE AIR.

Suppose the chimney to be in full draught, the hot air issuing at D with a velocity of, say, 100 feet per second, and let the heat be shut off. During the next second 100 cubic feet of hot air will escape at D and an equal volume will pass out of the cold-air shaft through the grating without undergoing any expansion, so that the upper surface of the air in the shaft must fall $\frac{1}{100}$ of a foot. Suppose, in the next place, that there is no draught in the chimney, a damper at C being closed, and let the grate be of such capacity as to hold 50 cubic feet of air, which we will suppose to be cold. Now close the circuit and heat the air until it expands from 50 to 100 cubic feet. This will raise the surface of the air at A $\frac{1}{200}$ of a foot.

When the chimney is in full draught and the heat supplied

continuously both of these processes are going on together. The surface at A is falling $\frac{1}{100}$ of a foot each second, on account of the flow of the air, and it is being lifted $\frac{1}{200}$ of a foot, by reason of the expansion of the air, the two operations being independent of each other.

The effect of heating the air is then simply to keep raising the cold air in AB, which acts as a weight to keep the apparatus running—*i.e.*, to keep the chimney drawing—one-half of this weight being balanced against the weight of the hot-air column, and the other half being employed in maintaining the draught.

In some clocks the operation of winding interferes with the running; in others it does not. I once arranged an astronomical clock in a manner corresponding so exactly in its principle with that of chimney draught that a description of it may advantageously be given. A (Fig. 140) is a pendulum clock. B is the main arbor to which the power is applied; on this arbor is a wheel, with a **V**-groove in its periphery, around which passes an endless cord. This cord, passing also around a similar wheel C free to rotate in the direction of the hands only, sustains a larger and a smaller weight, as shown. By rotating the wheel C the clock can be wound without interrupting the action of the weight upon B, whereas in an ordinary clock, with a cord simply wound on a drum at B, the winding up of the weight interrupts its pull on the shaft B and may cause the pendulum to lose a beat, or, if the winding is done very slowly, the clock may stop.

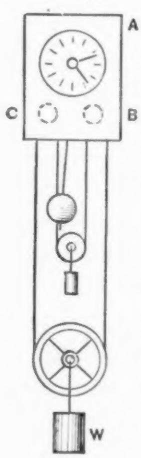


Fig. 140.

There is nothing to prevent the winding from being done so slowly as to become a continuous operation, the weight being lifted each second the exact distance that it falls, so that it remains stationary.

We may easily vary the construction in Fig. 139 so as visibly to separate the action of raising the weight from that of producing the velocity.

Suppose (Fig. 141) that the cold-air shaft be extended to twice its former height and connected at the top with a similar shaft GH, opening at the bottom with a grating of wires to heat the entering air.

The atmospheric pressure must be supposed the same at A as at D, and it could be made so by having a gasometer top AH with

sufficient weight to make up for the different actual barometric pressures at D and A.

In such an apparatus the hot grate would keep GH full of hot air, having substantially no velocity, the action of the heat being to make each cubic foot of cold air 2 feet high, or to expand a column half HG high into one with the height HG, the weight re-

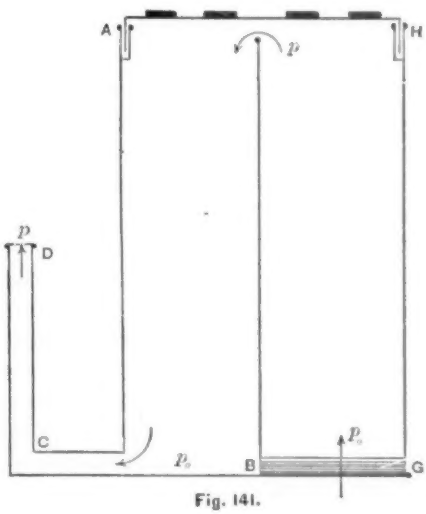


Fig. 141.

maining the same, so that the hot-air column, plus the pressure p, must balance the atmospheric pressure p_0 at the bottom.

At the top there would be a constant flow from H toward A, to keep the shaft AB full.

THE PRODUCTION OF THE VELOCITY.

The portion of the apparatus $ABCD$ would be devoted to producing the velocity of exit at D, and this velocity would evidently be that due to the height AD, or to the excess of the height of the column of hot air AB over that of a column of cold air of equal weight and section.

We shall have, then, *velocity* $= \sqrt{2g\,(AD)}$.

Were it advisable to take account of the change in density between the bottom and top of the columns of air, we should need a thermo-dynamic formula to express it exactly, but, as this whole change is not 1% in the height of the chimney, it may be neglected, or, if it be desired to take it into account, the density may be sup-

posed to vary regularly from top to bottom. When it is remembered that both the hot air in a chimney and the cold air outside increase in density downward, and that it is only the difference between these effects that influences the chimney's action, it will readily be seen that there is no use in attempting to include variations in density.

For the purpose of this paper it was desirable to assume the simplest conditions, and therefore to leave out friction and other complications. The necessary introduction of these by writers upon the subject in connection with actual chimneys in no way affects the argument of the paper showing the action to be a purely gravitational one. The introduction of an additional resistance simply requires an additional head to balance it—*i. e.*, an extra height of chimney.

It should be remembered that calculations of chimney draught, such as are to be found in the usual treatments of the subject, are made on the supposition that the chimney is the most contracted part of the air passage, as is usual where plants are properly proportioned. It is quite easy to vary the conditions sufficiently to render changes in the calculations necessary, but it would be useless to consider such cases here, because, were calculations of peculiar cases to be made, it would probably require as much judgment to know where to apply them as to make them. It will be sufficient to say that if the draught be throttled, "so to speak," by an extremely small grate or an impervious bed of coals, or by closing the ash-pit doors, we shall not have the usual conditions. At least it would then be quite out of place to calculate the velocity on the supposition that there was a chance for the chimney to act normally, or to discuss the temperature in the chimney for the production of the best draught. Naturally we should open the ash-pit doors and clean the fire before thinking it worth while to criticise the action of the chimney.

DISCUSSION.

Prof. H. B. Gale.—This paper, considered as a mathematical discussion of the theoretical flow of heated air in various forms of apparatus more or less resembling a chimney and furnace, possesses a considerable interest; but it will be admitted, I think, that it is necessary to assume rather too many peculiar conditions in the demonstration of these formulas to allow of their being safely applied to the conditions of practice.

One remark, made near the end of the paper, is worth noting, especially as several writers on this subject have made statements to the same effect, viz., that it is as useless to discuss the problem of chimney draught, whenever the draught is "throttled, so to speak," at the grate. To *throttle* the draught means to check its velocity by contracting the passage through which it flows. The evidences of throttling at any section of the passage are, first, a velocity in other parts less than would be attained with a uniform passage; second, a considerable difference between the pressures existing on the two sides of the throttled section. These effects are always found in ordinary boiler furnaces, and their amount can readily be ascertained by measurements with the ordinary siphon draught gauge. In this connection I may say that I have found it more convenient in making measurements upon chimney draught to use alcohol of known specific gravity in the draught gauge instead of water, as the indications with it are more sensitive.

It will appear upon investigation that a theory which applies only to cases where there is no throttling at the grate does not apply at all in ordinary practice. According to Rankine the throttling at the grate is sufficient, under average conditions, to cause a loss of head equal to three-fourths of the whole draught of the chimney. Of the remaining one-fourth, 75% goes to balance the frictional resistances of the flues and chimney, leaving only about one-sixteenth of the total head for "the production of the velocity," according to the laws deduced in this paper. In most cases in which I have made measurements of the velocities and pressures in existing plants, more than fifteen-sixteenths of the total draught has been expended in overcoming frictional resistances, and less than one-sixteenth in imparting energy of motion to the gases. For example, in the case of which the details are given in paper CCCLXXVI., on *Theory and Design of Chimneys*, in the present volume of the *Transactions*, only about $3\frac{4}{10}$% of the draught was needed to produce the velocity of discharge of the gas. In a boiler test made a few weeks ago, in which the mean velocity of the gases and the pressures were both carefully determined, about 60% of the whole head was lost by throttling at the grate, and the larger part of the remainder was absorbed by friction of the gases in the boiler tubes and chimney. In this case the height of the chimney was 92 feet and the temperature of the gases 609°. If the whole head had been employed in producing velocity of the gases, their velocity, by the formula deduced in this paper, would

have been about 78 feet per second. The mean velocity by measurement was only 16 feet per second, showing that only about 4% of the total head was expended in accelerating the gases.

The forces which balance or neutralize the greater part of the total head due to the draught of a chimney may be enumerated as follows, in the order of their relative importance: first, the resistance offered by the grate and fuel-bed; second, the resistance due to the boiler tubes (for small tubes this resistance is sometimes larger than that of the grate, but is usually less); third, the resistance due to friction in the flue and chimney. The small remaining head, unbalanced by resistances of this kind, is employed in various ways in imparting energy of motion to the gases. Part of this energy is represented in the velocity of the gases discharged from the top of the chimney; part of it is employed in accelerating the air entering the grate, this energy being mostly dissipated in heat when the small streams of air passing through the fuel emerge into the enlarged combustion chamber; energy is again absorbed in accelerating the gases entering the boiler tubes, to be again lost in heat when they emerge from the tubes into the uptake. Either of the two latter quantities of energy may be larger than that represented in the velocity of the gases discharging from the chimney; but all of them taken together are usually so small as to be insignificant in comparison to the energy absorbed in friction in the grate and boiler tubes.

All recorded experiments upon boiler chimneys agree in sustaining this view of the controlling importance of the frictional resistances. Now, what appears to me as chiefly open to criticism in this paper of Prof. Webb is that all this mathematics has been devoted to the calculation of the insignificant fraction of the total head which appears in the energy of the gases discharged from the top of the chimney, while the frictional resistances, which absorb the greater part of the head, have received no consideration.

Roughly speaking, what happens when a fire is kindled in a furnace is this: the heat causes a rarefaction of the gases in the chimney and a reduction of the internal pressure. The excess of pressure outside then accelerates the flow of air through the grate until *the friction*, increasing with the square of the velocity, balances, or nearly balances, the difference in weight of the outer and inner air columns. When this equilibrium is reached the velocity cannot increase further. The important practical problem to be solved, therefore, in the theory of chimney draught, is to find

an expression for the *frictional resistance* of the air *passing through the grate* in terms of its velocity. Then if we put this expression equal to the difference of weight of a unit column of gas inside the chimney and a similar column of the air outside, we have an equation which will give us roughly the velocity of flow through the grate. The velocity in the chimney will be greater or less than this, according as the ratio of grate opening to chimney area is greater or less. The *area of grate opening* is, therefore, an important factor, and should appear in a rational formula for chimney draught.

In the calculation as just described there would be an error of from 20% to 40%, due to the neglect of the resistances of boiler tubes and chimney. It is, however, not necessary to neglect these less important resistances, or even the small item of head due to the velocity of the escaping gases represented by $\frac{v^2}{2g}$ (or AD, in the paper before us); but it would certainly be much more reasonable to ignore the latter quantity and take account of the friction than to devote our energies exclusively to the relatively insignificant item of the head due to velocity in the chimney.

In regard to the calculation of these frictional resistances, I have proposed a method to which I think no valid objection has yet been raised, and which, though not professing to be more than a rough one, gives results which, in those cases where I have been able to test them, are not far from the truth; but it will be only after we get the real nature of the problem clearly before us that we can hope to establish, by experiments upon chimneys, more reliable constants.

Prof. Thomas Gray.—In the first paper, as I understand it, Professor Webb refers to the discussion of theoretical formulas for the velocity of air, one column of which is kept hot and the other cold. That discussion does not bear very directly on the practical problem of chimney draught, and beyond a single word or two I have nothing to say with reference to it. With regard to the actual chimney draught, besides the friction of the grate we have to bear in mind the fact that the gases passing through the chimney do not agree, either in composition or density, with the air outside. Consequently we have in the actual chimney a more complicated problem than that proposed for discussion by Professor Webb.

I have felt a little puzzled, in looking over Professor Webb's

second paper, by a statement near the beginning of it to the effect that this problem was purely a gravitational and not a thermo-dynamic problem. That does not seem to me to be quite the case. If Professor Webb simply means that we can calculate the velocity of efflux from the difference of "head," then I agree, and I believe we can do so to a close degree of approximation. When we consider, however, that we have practically here two columns of air, one of very large area, the other of limited area, and the small-area column raised to a higher temperature than the other, we find that we have simply a circulation. We have no change of potential energy produced, because the loss in the cold column is compensated for by the gain in the hot column. The height of the centre of gravity of the whole mass remains practically the same. We must look to something else, then, for the source of energy in the flow, and that source of energy is evidently in the heat supplied at the point where the temperature changes. I should say, therefore, that the problem is fundamentally a thermo-dynamic and in only a secondary sense a gravitational problem.

Prof. Webb.—I wish Professor Gray would state more succinctly why he considers it a thermo-dynamic problem.

Prof. Gray.—Simply because of the fact that we have to look to the expenditure of heat as the motive power in the operation.

Mr. Nagle.—I would like to ask Professor Gale why he prefers alcohol to water in measuring the height of the column in the chimney.

Prof. Gale.—The alcohol flows more freely, is more sensitive, and has a smaller specific gravity, which gives a greater indication for the same pressure.

It evaporates, of course, but that does not affect it particularly.

Prof. Webb.—Will Professor Gale be so good as to illustrate his method of measuring the pressure in the chimney?

Prof. Gale.—The apparatus that I used is simply the ordinary siphon gauge. It usually is filled with water, though I have generally preferred to substitute alcohol. To obtain the difference in pressure between the air under the grate and in the bottom of the chimney, or the fall of pressure during the passage of the gas through the furnace as a whole, you would simply connect the siphon to the bottom of the chimney, leaving the other end open to the air. Then the difference of level in the liquid in the two branches of the U-tube would give us the difference between the air under the grate and the gas at the bottom of the chimney. To get the fall

of pressure in the grate, connect the siphon with the fire box just above the grate; then you get the difference of pressure between the air under the grate and the air above the grate, or the fall, as the air passes through the grate. A perfectly plain glass tube is used, having the two branches close together, with only a finely divided scale between them, so that accurate differences of level may be obtained. I usually found it best to leave only a very small opening in one end of the tube, so that the vibration might be checked a little.

Prof. Webb.—What about the inner end of the tube? Has it a straight end?

Prof. Gale.—I have generally used an iron pipe with a bent end running into the chimney and connected the siphon to it with a rubber tube; then by turning the pipe around, the end can be pointed in different directions, and if there is any difference fact due to the different directions of the opening, that can be averaged up.

Prof. Webb.—Probably in different places in the chimney you would find different pressures.

Prof. Gale.—Yes, sir; slightly different. There is usually no perceptible difference in the section across, but turning the open end of the pipe in different directions changes the reading slightly. The effect of that is usually very small, however. All that I have usually done is to use a bent tube and place it in different parts of the cross-section, turning it in different ways, and average the readings if there was a difference.

Prof. W. T. Magruder.—I would like to ask if any member present has any data which he has taken as to the pressure at the bottom of the chimney, and also at the top of the chimney at the same time, and also in the centre of the chimney *versus* the edges.

Prof. Gale.—If I might be allowed a word about that, I have had no experience, but I have a plan, which it might be interesting to the members to see, for doing that—whenever I am able to carry it out. I can illustrate it in just a moment. The plan would be to carry a small iron pipe from the top of the chimney down through the inside and connect the bottom of this pipe to one end of your siphon gauge, while the other end is in communication with the space at the bottom of the chimney. By means of having the tube running down through the chimney, the temperature of the gas in the tube would be the same as that of the gas in the chimney around it, and the weight of the column in the tube

would be the same as the weight of an equal column of gas outside. In that way you get the difference in pressure to balance the frictional resistance in the chimney itself, or the pressure which is unbalanced by the weight of the gas. I have never been able to carry out that plan as yet.

Mr. Borden.—Mr. Barrus, will you not describe the gauge that you use for a draught gauge?

Mr. Barrus.—The apparatus to which Mr. Borden refers consists of a plain glass U-tube, represented at *A* (Fig. 177), with two large chambers at the top. The level of the liquid is carried in these chambers, and a slight movement in either chamber produces a very large movement of the liquid in the tube. I use in the left-hand chamber a kind of oil which is called Downer's heavy spindle oil, and in the right-hand chamber alcohol, which is colored by any simple means. Each extends, on its respective side, down to the point *C*, where there is a distinct line of demarcation between the alcohol and the oil. Now, when the right-hand chamber is connected with the chimney the suction of the draught produces a difference of level in the chambers of, say, $\frac{1}{2}$ inch. In proportion as the area of the chamber is to that of the tube the liquid in the tube is drawn down a greater distance. If the chamber is 2 inches in diameter and the tube [1] inch, which are about the sizes used, the distance would be something like eight times.

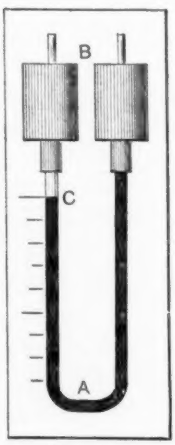


FIG. 177.

Prof. Gale.—Would a correction come in there for the difference of specific gravity?

Mr. Barrus.—There is a correction for that, and I have generally made it in this way: I have connected the apparatus with an ordinary U-tube containing water, calibrating it by this means. I find with this instrument that there is no difficulty in getting a correct indication of very slight amounts of draught.

Prof. J. B. Webb.—I asked to have this paper and the preceding one discussed separately, as they are quite dissimilar, and as it was so ordered and a discussion upon that one was duly announced by the president, there would be no occasion to refer to it here had it not been commented upon by Prof. Gray. With this exception I do not think that any of the discussion touches it.

It seems necessary to state again what the object of that paper is, having evidently failed to make it sufficiently clear in what was said when introducing it. It simply takes a treatment contained in the last edition of Péclet and shows it to be unnecessarily long, and that although thermo-dynamic formulas are introduced, they are eliminated before the close of the demonstration. It shows also that by making simple changes the argument runs into the ordinary one given by those who do not pretend to use thermo-dynamics. The paper does not depart from its object to discuss the applicability or non-applicability of the hypotheses and resulting formulas to this or that chimney, and it ought not to. It is enough to add that if any one will examine Péclet's treatment and read the paper carefully, it will be seen that what has been said has no legitimate bearing upon it. If any one wishes to place himself on record as saying that Péclet's "does not bear very directly on the practical problem of chimney draught," I have no objection, but the natural inference must be that such critic is not familiar with the treatment of the subject by Péclet and others. This view is further justified by the statement that in "actual chimney draught" there is "friction of the grate" and "the gases do not agree either in composition or density with the air," just as if Péclet, after completing the discussion of a ventilating flue, did not immediately proceed to modify the formulas so as to include all these complications in an ordinary chimney.

To come now to the criticisms of my second paper, which is legitimately under discussion, Prof. Gray says that he was puzzled by my statement that "the problem is not a thermo-dynamic one," but thinks that after all it is, and gives as his reason for thinking so "the fact that the expenditure of heat is the motive power in the operation."

In reply I have three things to say:

1. At the most it is simply a question as to the correctness of a term employed, and beyond this has no effect on the paper.

2. I believe my use of the term thermo-dynamic to be justifiable, because I made the statement first in reference to Péclet's treatment, contained in the discussion of a paper read at the last meeting, and in this treatment the term is employed to distinguish a part of the discussion involving the use of the formulas for adiabatic expansion from a previous part depending upon the law of Charles, which was known before the science of thermo-dynamics existed.

3. I think my use of this term is correct, as it certainly agrees with its use by Rankine and other leading writers. As the air goes up the chimney it expands adiabatically, to be sure, on account of the decrease of pressure; but as this expansion furnishes less than two-tenths of a per cent. of the whole energy, it has very little to do with the action of the chimney, and is always, I believe, neglected. The main action is gravitational, and the argument to the contrary that the "height of the centre of gravity of the whole mass remains practically the same" is misleading. What bearing the "very large area" of one air column and the "limited area" of the other have upon the question the professor does not state.

Prof. Gray.—Where does the kinetic energy come from?

Prof. Webb.—It becomes kinetic by the action of gravity; without gravity there would be no such thing as chimney draught.

Prof. Gray.—It cannot do that because the gas rises up and keeps the centre of gravity the same. We have, however, the expenditure of heat. We take energy from the heat, and so long as we use energy derived from heat we have thermo-dynamics.

Prof. Webb.—I knew that the question might, as intimated by the professor in the first part of his criticism, be a puzzling one, and hoped that the paper, if carefully read, might make the matter clear; in this instance it seems either not to have been so read or to have failed to convince. It may therefore not be amiss to add another illustration.

Suppose a company engaged in the sale of water-wheels to have a mill-pond where they exhibit their turbines, etc., in operation. They experiment upon and discuss the performance of these wheels in accordance with the usual laws of hydro-dynamics, which regard the action as a gravitational one. Suppose now that some one should become impressed with the fact that the water in the pond remains at a constant level, all the water that is used being afterward evaporated by the sun and returned to the pond as rain, and that consequently the energy given out by the wheel is derived ultimately from the heat of the sun: would such a view of the case be any justification for a claim that the problem was not a hydro-dynamic but a thermo-dynamic one?

Or suppose, even, that in a time of drought a steam-pump were set up to keep the pond full of water, so that without leaving the premises the whole circulation of the water could be traced, and it was at once evident that the centre of gravity of the whole

mass of water remained at a fixed height: would the problem of the performance of the water-wheels be any less a gravitational one and more of a thermo-dynamic question than before?

I doubt if any one could seriously entertain such a view. Almost all the energy we have comes or has come from the sun, but all problems of energy are not thermo-dynamic ones.

The ordinary inclined-track horse-power is also a case in point. How the horse develops the energy he furnishes has nothing to do with the action of the machine, which is the same whether the horse works in the usual way or stands still, and goes down hill as a simple weight would. The energy furnished by the horse-power is due to the weight of the horse falling as fast as it would fall if he stood still on the inclined track. The problem is a gravitational one.

With regard to the discussion of the other participant in the debate, a careful examination fails to show that it has more than a nominal bearing upon the matter of the paper.

After a complimentary remark the first paragraph concluded with a statement unintelligible as applied to my paper. Said paper contains but one formula, which formula I have not attempted to demonstrate, as it is too simple and well known, and far from its involving "many peculiar conditions," none but the simplest have been assumed. The formula is, furthermore, perfectly applicable to all the conditions of practice, and is the one used by the critic himself.

The next two paragraphs are based upon a misstatement of the concluding remarks of the paper, and claim, therefore, no serious notice. I have not said therein that it is useless to discuss the problem of chimney draught when throttled by an ordinary grate and bed of coals, but that unusual conditions necessitate changes in the calculations, and that it would be useless to attempt "to consider such cases *here*"—*i. e.*, in a short paper devoted to a different object. Certainly when there is throttling "by an extremely small grate or an impervious bed of coals, or by closing the ash-pit doors, we shall," as I claim, "not have the usual conditions."

Two-thirds of the way along it is stated, "as chiefly open to criticism," that "all this mathematics has been devoted to the calculation of the insignificant fraction of the total head which appears in the energy of the gases discharged," "while the frictional resistances, which absorb the greater part," "have received

no consideration." This is equivalent to a statement that there is no criticism at all, for it is precisely the purpose of the paper to discuss the true nature of the head which produces the velocity in a flue with no other resistances, and, by inference, the true nature of the head in any chimney where a part thereof, no matter how large, may be balanced by resistances.

The frictional resistance in a chimney is, no doubt, a legitimate subject for a paper, but the present paper is not upon that subject. It is somewhat discouraging to a mathematician, when he has exerted himself to keep mathematics substantially out of a paper and has flattered himself on his success in producing one with but a single formula in it, to have it referred to as "all this mathematics;" and, further, it is not correct to state that the paper attempts to calculate the "insignificant fraction" of head producing the velocity of discharge in an ordinary chimney. There is no such attempt and any figures given are purely illustrative.

I asked for a more particular description of the speaker's method of measuring the pressures in the chimney, not at all for the purpose of getting a description of an ordi ary gauge and its use in showing the considerable fall of pressure due to a bed of coals, which must be well known, but to see if there had been any attempt to make nice measurements. There seems to have been none; no distinction seems to be made between atmospheric pressure and the pressure in the ash pit, and the siphon gauge has been but slightly improved by using alcohol instead of water. I do not suppose the speaker means that the former fluid is more "convenient" than the latter, but that it gives a somewhat larger indication for the same pressure. What is meant, too, by flowing "more freely" and being "more sensitive" is not quite clear, for when a small hole has to be interposed to prevent sudden fluctuations, one would suppose water to be sufficiently sensitive. Now, if the total head is, say, one inch of water and fifteen-sixteenths is used at the grate, something more delicate than an ordinary siphon gauge will be needed to investigate the distribution of the remaining sixteenth. There seems to be no acquaintance with various draught gauges for the indication of small differences of pressure. I have no doubt that Mr. Barrus' will give good results when properly standardized, and I suppose the standardizing allows for any capillary action in the small tube. Mr. Kent described to me a similar device of his some years since. The principle is the same as Mr. Barrus', except that but one liquid

(water) is used and a small bubble of air takes the place of the visible junction of the oil and alcohol. This bubble has a play of several inches in a small horizontal tube connecting two upright tubes of much larger diameter, and the only standardizing needed is to ascertain the ratio of the cross-sections of the large and small tubes. The displacement of the bubble will, of course, be greater, in just this ratio, than the change of level in the large tubes. At the Electrical Exhibition in Philadelphia I devised a simple gauge which was used in nearly all the tests and in other tests since, and which is described in the report.

Fig. 230 illustrates this gauge. S is a pair of ordinary scales capable of weighing to any desirable fraction of an ounce. The rest of the apparatus is shown in section as follows: B is a flat

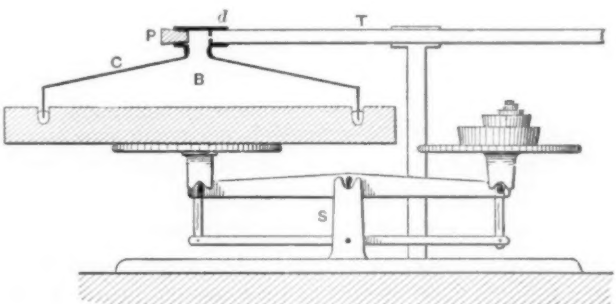


Fig. 230.

board in which a circular groove is turned and nearly filled with mercury. This board rests upon the scales. C is a sheet-metal cover whose lower edge dips into the mercury. This edge should be quite thin and, if necessary, protected from the mercury by varnish; but it is better to make it of thin sheet-iron with a riveted joint made tight with varnish. The inside of this cover is connected with the place in which the pressure is to be measured by means of the tube T, suitably supported by an upright from the base of the apparatus. The end of T is almost stopped by a diaphragm d with but a small hole in it, so that while no rapid fluctuations of pressure can reach the inside of B, the same average pressure will be maintained there as is to be measured. P is a large plug or cork; when it is withdrawn it places the inside of B substantially in communication with the atmosphere, notwithstanding the small hole at d, which may remain open.

The scales can then be balanced, and upon the plug being replaced they will show a difference of weight equal to the change of pressure inside of B, multiplied by the surface over which it acts. Thus, if the pressure to be measured is, say, 0.04 ounce below atmospheric pressure (about one-sixteenth of an inch of water), and if the cover is scant 8 inches in diameter so as to cover 50 square inches of the board, the reduction of pressure on the board, due to the insertion of the plug and the lowering of the pressure within the cover by 0.04 ounce per square inch, will be $50 \times 0.04 = 2$ oz. With sensitive scales protected from currents of air such an apparatus will give reliable indications of extremely small differences of pressure.

Professor Lanza's adaptation of a well-known principle to the construction of such a gauge was also used in some of the tests made at the same exhibition.

Why so much space should be devoted to emphasizing the fact that a large part of the total head is needed to get the air through the bed of coals, a fact which can scarcely be new to any one who knows anything of the subject, is unexplainable to me. The elaborate description, moreover, of what goes on inside of a furnace, while it would require but a slight knowledge of mechanics and some imagination for its production, would need a much deeper knowledge and very careful experiments for its verification. Indeed, there seems to be some confusion in the statement itself between the purely frictional and the other resistances, and while the fact that the area of the grate opening has to do with the problem has not, to my knowledge, been denied, its importance is not made any clearer by the loose and incorrect statement: "The velocity in the chimney will be greater or less than the velocity through the grate, according as the ratio of grate opening to chimney area is greater or less." Greater or less than what?

I believe it to be a very important duty of the mathematician to explain and illustrate thoroughly the meaning and application of some of the simpler formulas and phenomena of mechanics and physics; and in my paper I have tried to perform that duty with regard to the phenomenon of the production of the velocity of the gases in a ventilating flue or a chimney, and I know that such explanations as I have given have put the matter in a clearer light to others. I differ entirely from Professor Gale in his estimate of the relative importance of, say, the fifteen-sixteenths of

the head required at the grate and the one-sixteenth needed for the velocity of exit. The whole object of the chimney being to produce that velocity, I cannot agree with him that it would be "more reasonable to ignore" it, "take account of the friction than," etc. In fact, the frictional loss is simply an expense attendant on the production of the absolutely necessary velocity.

CCCLXXXVII.

A UNIVERSAL STEAM CALORIMETER.

BY GEO. H. BARRUS, BOSTON, MASS.

(Member of the Society.)

THE subject of steam calorimeters has, in various ways, been brought to the attention of the Society so many times that the members may be tired of hearing anything more about it. What has heretofore been said has not, however, exhausted the subject, and the author may be pardoned for bringing the matter to notice again in the present paper.

In one of the discussions* at the last meeting the author introduced a brief notice of what he has named "A Universal Calorimeter," and he made the statement that at a future meeting he would describe the instrument more in detail, and give the results of some experimental work upon it, in connection with evaporative trials of boilers, and this he now proposes to do.

Besides this, a special investigation has been carried out, the object of which was to determine the accuracy of the instrument, and an account of this will be added.

In experimenting with the author's superheating calorimeter,† where the quality of steam was being tried, which was very wet and almost beyond the range of the apparatus, a device was planned, though never perfected, for passing the steam first through a chamber in which some of the moisture would be deposited, and thus relieve the instrument from handling so much water, thereby increasing its range. It may be mentioned that the use of such a device in connection with the superheating form of instrument will enable any desired amount of moisture to be measured by that instrument and overcome the objection of limited range which has been raised against it.

This method of treating a part of the moisture in wet steam

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* *Transactions*, A. S. M. E., Vol. XI., page 205.

† *Transactions*, A. S. M. E., Vol. VII., page 178; Vol. VIII., page 235.

where the quantity is excessive has been perfected in the apparatus now brought to notice, but in place of using a superheating calorimeter for determining the remaining quantity of moisture, recourse has been had to the wire-drawing principle, the application of which to calorimeter work formed the subject of Professor Peabody's paper at the Scranton meeting.*

The appended cut (Fig. 142), reproduced from the report of the discussion referred to, shows the apparatus. The principal parts of the instrument consist of the chamber *A*, or "drip-box," as it is called, and the "heat-gauge," consisting of the orifice *I*, and the

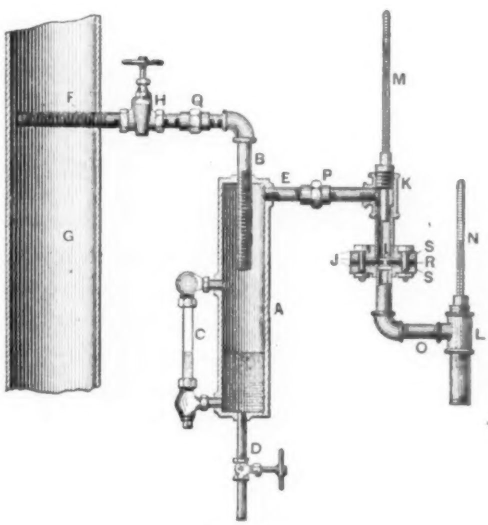


Fig. 142.

two thermometers *M* and *N*. The instrument is connected to the main steam-pipe *G*, which carries the steam to be tested, by means of the perforated pipe *F*, and this pipe extends across the full diameter, in order to obtain a sample of the steam tested. The orifice *I* opens into a pipe which is in free communication with the atmosphere. By the use of the orifice a continuous current of steam is made to pass through the whole apparatus, and the current has a constant rate so long as the pressure is constant. In the form thus far made, the supply-pipe *F*, and the fittings, up to the drip-box, are the ordinary size of ½-inch steam-pipe. The drip-box is 2 inches

* *Transactions*, A. S. M. E., Vol. X., page 327, Paper No. 323.

inside diameter and 12 inches long, and the drain-pipe D is $\frac{1}{4}$-inch pipe. The pipe leaving the drip-box is also $\frac{1}{2}$-inch, and the remaining pipes and fittings are of the $\frac{3}{4}$-inch size. The parts marked S, which enclose a plate in which the orifice is placed, are a pair of union flanges, and inserted between these flanges and under the bolt-heads are pieces of non-conducting material which prevent the direct transfer of heat through the metal walls of the high-pressure pipe above the orifice to the walls of the low-pressure pipe below it. The thermometer M rests in an oil-cup, K, as shown, and the thermometer N is arranged in a like manner.

The use of the non-conducting material shown at the points J might seem to some unnecessary, and it may be explained that in some early experiments with the superheating calorimeter it was found absolutely necessary to cut off all solid metallic connection between the jacket and the interior heating pipe, else there would be a transfer of heat from one to the other, which would make the indications of the thermometers erroneous. The same principle applies here, and justifies the arrangement which has been adopted.

The orifice I is made about $\frac{1}{8}$ of an inch in diameter for pressures in the neighborhood of 80 lbs., and under 80 lbs. pressure it discharges about 60 lbs. weight of steam per hour.

The range of the wire-drawing part of the instrument, or "heat-gauge," as it has been named, is a percentage of moisture varying according to the pressure. When this is 80 lbs., the range is between 3% and 4%. It is unnecessary to use the drip-box unless the quantity of moisture is in excess of, say, 3%. The unions P and Q are therefore made interchangeable. When a test is to be made, the heat-gauge is first applied directly to the union Q and a preliminary trial made, to see what the general condition of the steam is. Whenever the moisture exceeds 3%, or the limiting quantity at the existing pressure, the thermometer N shows a temperature of about 213°, and drops of water will generally be seen escaping from the open discharge-pipe. If the quantity of moisture is not beyond the range of the wire-drawing instrument, the temperature shown by thermometer N will be in excess of 213°.

It is generally advisable to bring the complete apparatus into use, if the thermometer N shows less than 220°. It is usually found, when the steam is wet enough to show this latter temperature, that the quantity of moisture varies, and thermometer N fluctuates over a considerable range, and at times it will drop to its limit of

213°. Whenever, therefore, thermometer N shows a temperature of 213°, either continuously or periodically, it is necessary to bring the drip-box into use.

In using the complete apparatus, the condensed water from the drip-box is drawn off, by means of the valve D, into a bucket resting on scales, and the quantity drawn off is regulated so as to keep the water level, as shown in the glass C, at a constant point. When the drip-box is used in this way, the author has found that almost the whole quantity of moisture in a sample will be deposited here, and very little moisture will be left to pass over into the heat-gauge. Indeed, the experiments show that the drip-box alone, with a suitable orifice or valve provided at the top, so as to obtain a proper circulation through it, would form a very satisfactory instrument for determining the quantity of moisture in any case where the steam contained much of it.

When the quantity of moisture drawn off from the drain-valve D has been determined for a given time, the percentage of moisture which this represents must be found by comparing it with the total amount of steam passing through the apparatus. The total may be determined either by computation or by trial. The computation may be made by finding the exact area of the orifice, and computing the quantity which passes through by means of the formula,

$$Q = \frac{\textit{Pressure above zero} \times \textit{area}}{70},$$

which gives the number of pounds discharged through the orifice per second. The pressure to be used is that corresponding to the temperature shown by thermometer M. The quantity, as thus found, is accurate enough for rough comparisons. The exact quantity can be determined by conducting the steam discharged from the open end of the apparatus into a tub of water placed on scales—or, what is a better way, into a coil of lead pipe or iron pipe surrounded by flowing water, in the manner of a surface condenser, and weighing the condensed water drawn off in a given time.

A certain amount of moisture is produced by radiation from the apparatus itself, even though all the parts are well covered, as it is quite necessary that they should be, with hair felting. The readings of the instrument on the test must therefore be corrected for the loss thus occasioned. It has been the practice of the

author to make these corrections by observing the indications when the apparatus is supplied with steam from the pipe G, at a time when the pressure is steady and the pipe contains nothing but dead steam, there being no current. This condition of things can generally be obtained in a factory at noon-time, when the engine is stopped, or at night, after the close of the day's work. It may fairly be presumed that the apparatus is then supplied with dry steam, and whatever moisture collects in the drip-box A, and whatever difference is shown by thermometers M and N, is due simply to the loss of heat from radiation. When the loss from radiation has been thus obtained, the quantity representing that due to the drip-box is simply subtracted from the weight of water drawn off during the same length of time on the main test. The way in which the correction is applied to the readings of thermometers M and N is to take the reading of thermometer N on the radiation test when thermometer M indicates an average, and use this reading as a starting-point. The indication of thermometer N on the main test is then simply subtracted from this normal reading. For example, suppose the average reading of the upper thermometer (M) during the main test is 312°; suppose the lower thermometer (N) indicates an average of 260° on the main test, and on the radiation trial suppose it indicates 267° when thermometer M shows 312°; the process would be simply to subtract 260 from 267, and this would give as a result 7° as the cooling effect produced by the moist steam discharged on the main test.

In order to compute the amount of moisture from the loss of temperature shown by the heat-gauge, the number of degrees of cooling of the lower thermometer (N) is divided by a certain coefficient, representing the number of degrees of cooling due to 1% of moisture. This coefficient depends upon the specific heat of superheated steam, which, according to Regnault's experiments, is 0.48. In other words, the heat represented by 1° of superheating is 0.48 of a thermal unit. The author's experiments show that this quantity cannot be applied exactly to the form of instrument under consideration. The quantity to be used varies somewhat according to the degree of moisture. For an instrument working under a temperature of 314°, by the upper thermometer, and with a cooling by the lower thermometer from 268° to 241°, the quantity was found to be about 0.42. When the cooling, however, was from 266° to 225°, the quantity to be

used was found to be about 0.51. The experiments have not as yet covered a sufficient range to determine the exact law which can be applied to every case, but it seems probable that the specific heat is more or less constant until the temperature by the lower thermometer approaches the point of saturation for the low-pressure steam, while beyond this point the specific heat rapidly increases. For the present, it is assumed that the quantity 0.42 is the proper one to apply whenever the temperature by the lower thermometer is above 235°, and that in cases where the temperature is below 235°, the quantity to be used is an increasing one, reaching perhaps to 0.55 when the temperature drops to 220°.

One per cent. of moisture, now, represents the quantity of heat determined by multiplying the latent heat of one pound of steam, having a pressure corresponding to the indication of thermometer *M*, by 0.01, and this product is to be divided by 0.42 (provided the lower temperature is not below 235°), in order to express it in terms of degrees of superheat. For example: When thermometer *M* shows 312°, the latent heat is 894 thermal units, and 1% of this is 8.94; dividing by 0.42, the number of degrees of superheat corresponding to 1% of moisture is found to be 21.3. For several other temperatures, which cover the ordinary range that would commonly be used, the necessary coefficient is given in the following table:

Temperature by Thermometer *M*.	Coefficient.	Temperature by Thermometer *M*.	Coefficient.
270	22.0	320	21.1
280	21.8	330	21.0
290	21.7	340	20.8
300	21.5	350	20.6
310	21.3	360	20.5

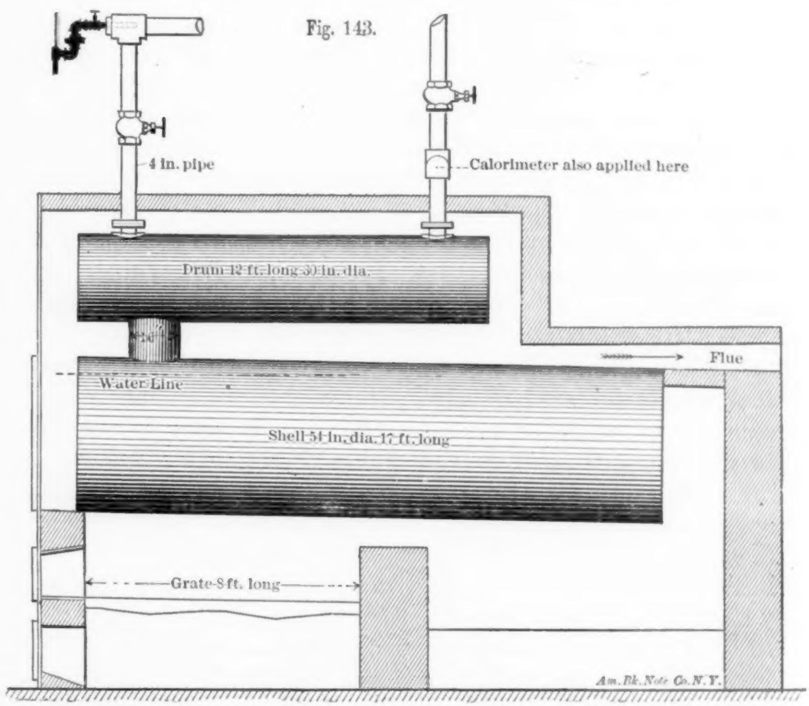
The utility of the instrument, and the general manner in which it operates in practice, may best be shown by reference to the various tables which are appended, giving the results of a number of tests on different boilers, made by the author.

TEST NO. 1.

The first test was made when using simply the wire-drawing part of the instrument, or "heat-gauge," in a somewhat incom-

plete form. Only one thermometer was applied—that is, the lower thermometer—and the temperature of the steam above the orifice is only to be learned by the indications of the pressure-gauge attached to the boiler. The boiler was one which gave steam of varying degrees of dryness, and it was admirably adapted for a test of this kind. It consisted of two shells, as shown in the appended sketch (Fig. 143), the lower one of which was nearly filled with tubes, and the upper one served the purpose of a drum.

Fig. 143.



The connection between the two shells was by a single neck at the front end. Two steam-pipes carried away the supply of steam, one being attached at the front end of the drum, directly over the connecting neck, and the other being attached to the rear end. The smoke and products of combustion, on leaving the tubes, passed over the exterior surface of the drum on their way to the chimney, and the drum thus furnished a certain amount of steam-heating surface. The calorimeter was attached first to one of these pipes and then to the other, and the form of instru-

ment used, as also the method of attachment, is shown in the sketch. The boilers at this place were two in number, and the calorimeter was attached to only one. When the instrument was applied to the front pipe, the quality of the steam indicated was quite variable; but when it was attached to the rear pipe, the quality became nearly constant, and the indications pointed to a small amount of superheating. The indications of the instrument on the front pipe revealed an exceedingly interesting state of affairs. The quality of the steam fluctuated periodically over a considerable range, varying from a condition of extreme wetness to a nearly dry condition. When the indications pointed to the largest amount of moisture, the thermometer showed about 213°, and drops of water issued with the steam at the point of discharge. The periodical fluctuations were surprising, until a careful observation of the times when they occurred showed that the wet steam was produced according to the rate of production of steam in the boiler to which the calorimeter was attached. The firing of the two boilers was done alternately, and when the boiler to which the calorimeter was attached was fired, there was a cessation in the generation of steam in this boiler, and the other boiler, in which the fire was active, was drawn upon to make up the deficiency. At this time the thermometer in the calorimeter boiler always showed increasing indications. Whenever, on the contrary, the other boiler was fired, and at times when the fresh fires in the calorimeter boiler had become active again, the indications of the thermometer would rapidly fall. At such times the auxiliary boiler became, for the moment, inoperative, and the calorimeter boiler was called upon to furnish the greater part of the steam. The lower shell of the boiler being nearly filled with tubes, and the generating surface in this shell being very small, there was an active tendency for this shell to carry up into the drum a mixture of water and steam, and this would easily find its way into the front pipe, directly above, whenever the boiler was doing much work. The alternate heating and cooling, due to the periodical firing, produced alternately very wet and nearly dry steam in the front part of the drum, and, as a consequence, it produced a varying degree of moisture in the steam which passed through the front pipe, as noted.

DIMENSIONS AND OTHER DATA REGARDING BOILER REFERRED TO IN FIG. 143, TEST NO. 1.

1. Diameter of main shell....................................54 in.
2. Length of main shell and length of tubes....................17 ft.
3. Number of tubes, 4 in. outside diameter......................45
4. Size of grate....................................4 x 8 ft.
5. Area of grate surface....................................32 sq. ft.
6. Area of water-heating surface..........................934 "
7. Area of steam-heating surface..........................107 "
8. Area of total heating surface..........................1,041 "
9. Collective area through tubes..........................3.4 "
10. Ratio of total heating surface to grate surface.............32.5 to 1
11. Ratio of grate surface to tube area......................9.3 to 1
12. Kind of coal used................................Schuylkill pea
13. Percentage of ashes..17.8
14. Coal burned per hour per sq. ft. of grate..................11.5 lbs.
15. Water evaporated per sq. ft. of heating surface per hour... 3.1 "
16. Average temperature of flue gases..........................420°
17. Average draught suction....................................0.24 in.
18. Water evaporated per pound of combustible from and at 212°, uncorrected for moisture....................11.07 lbs.

TABLE No. 1*a*.

DOUBLE-DECK HORIZONTAL RETURN TUBULAR BOILER (Fig. 143). CALORIMETER ATTACHED TO FRONT STEAM-PIPE.

Time.	Boiler Gauge.	Temperature shown by Lower Thermometer of Calorimeter.	Condition of Steam at Outlet of Calorimeter as it Appeared to the Eye.	Position of Damper.	Time of Firing.	
					Auxiliary Boiler.	Boiler to which Calorimeter was Applied.
10.00	......	236	Dry	Wide open	10.00	....
10.01	......	226	Wet *	"	....	....
10.02	......	242	Dry	"	....	10.02¼
10.03	......	256	"	"	....	....
10.04	......	264	"	"	....	....
10.05	......	269	"	"	....	....
10.06	......	271.5	"	"	....	....
10.07	......	273.5	"	⅓ open	....	....
10.08	......	272	"	"	....	....
10.09	......	265.5	"	"	....	....
10.10	......	264.5	"	"	....	....
10.11	......	230	"	"	10.11⅓	....
10.12	......	218	Wet	Wide open	....	....
10.13	......	222	Dry	"	....	....
10.14	......	241	"	"	....	....
10.15	......	256	"	⅔ open	....	10.15
10.16	......	268	"	"	....	....
10.20	88	258	"	½ open	....	....
10.21	......	246	"	"	....	....
10.22	......	226	"	"	....	....

* By the term "wet" is meant that drops of water emerged from the outlet of the calorimeter.

TABLE No. 1*a*.—*Concluded.*

Time.	Boiler Gauge.	Temperature shown by Lower Thermometer of Calorimeter.	Condition of Steam at Outlet of Calorimeter as it Appeared to the Eye.	Position of Damper.	Time of Firing. Auxiliary Boiler.	Time of Firing. Boiler to which Calorimeter was Applied.
10.23	......	220.5	Wet	½ open	10.22$\frac{2}{3}$	....
10.24	......	215.5	"	Wide open	....	....
10.25	......	230	"	"	....	....
10.26	......	246.5	Dry	"	....	....
10.27	......	256.5	"	"	....	10.27
10.28	83	264	"	"	....	....
10.29	......	269	"	½ open	....	....
10.30	......	270	"	"	....	....
10.31	......	262	"	"	....	....
10.32	......	241	"	"	....	....
10.33	......	221	"	"	10.33	....
10.34	......	214.5	Wet	"	....	....
10.35	......	225	"	Wide open	....	....
10.36	......	242	Dry	"	....	....
10.38	87	261	"	"	....	....
10.39	......	257.5	"	½ open	....	....
10.40	......	255.5	"	"	....	....
10.41	......	259	"	"	....	10.41
10.42	......	265	"	"	....	....
10.43	......	269.5	"	"	....	....
10.44	......	272	"	"	....	....
10.45	......	274	"	"	....	....
10.46	......	273	"	Wide open	....	....
10.47	......	253	"	"	....	....
10.48	......	227	"	½ open	....	....
10.49	......	223.5	"	"	....	....
10.50	......	235	"	"	....	....
10.51	......	241.5	"	"	10.51	....
10.52	......	236	"	"	....	....
10.53	......	237	"	Wide open	....	....
10.54	87	245	"	"	....	....
10.55	......	251.5	"	½ open	....	10.55$\frac{1}{4}$
10.56	......	258	"	"	....	....
10.57	......	263	"	Wide open	....	....
10.58	......	267.5	"	"	....	....
10.59	......	265	"	"	....	....
11 00	......	235	Wet	"	11.00	....
Normal.	85	288 (Rear pipe.)		....	....	....

In Table No. 1*a* the indications of the thermometer, as also data regarding the condition of the issuing steam as it appeared to the eye, the position of the damper, and the time of firing of each boiler, are given for nearly every minute during an hour's test. The fluctuating character of the readings, as influenced by the time of firing, is clearly shown in this record. Notice, for example, the reading at 10.27, when the boiler to which the calorim-

eter was applied was fired with fresh coal; the reading is 256.5°, and, compared with the previous readings, the indication of the thermometer is rapidly rising; at 10.28 it reaches 264°, and at 10.30 it is 270°. At this time—three minutes after firing—it is evident that the boiler began to recover its normal rate of production, and the indications of the thermometer began to fall; at 10.33 they had dropped to 221°, and at this time the auxiliary boiler was fired, which occurrence threw nearly the whole work of production upon the calorimeter boiler, and this was followed by the thermometer going down to 214.5° and the issuing steam presenting a wet appearance to the eye. The thermometer then began to rise, and at 10.38, five minutes afterward, it had reached 261°, and it is presumed that the fire in the auxiliary boiler had become quite active, while that in the calorimeter boiler was somewhat cooled.

The normal reading of the instrument was not determined when applied to the front pipe, but, taking the indication for the normal as determined for the rear pipe, which was 288°, the cooling effect due to moisture for the best indication—viz., 274° at 10.45—is $288° - 274° = 14°$; and this divided by the proper coefficient—viz., 21—gives for the percentage of moisture, under these circumstances, 0.66. The lowest indication—viz., 214.5° at 10.34—shows a cooling effect due to moisture of $288° - 214.5° = 73.5°$; and this, divided by the assumed coefficient for this temperature, which is 16.9, gives, for the percentage of moisture, 4.34%. This percentage, however, does not show the whole for this particular case, because the range of the instrument was evidently exceeded.

The record of the test when the calorimeter was applied to the rear pipe is given in Table No. 1*b*. The indications of the thermometer vary from 288° to 297.5°, with a normal reading of 288° at 85 lbs. pressure. There is continual evidence here of superheating, and this might be expected from the fact of the steam-heating surface, of which the steam issuing from this end of the drum had the benefit.

TABLE No. 1b.

DOUBLE-DECK HORIZONTAL RETURN TUBULAR BOILER (same as preceding).
CALORIMETER ATTACHED TO REAR STEAM-PIPE.

Time.	Boiler Gauge.	Temperature shown by Lower Thermometer of Calorimeter.
1.00	84	291.5
1.01	82.5	290.5
1.07	72	288
1.11	73	287.5
1 19	69	289.5
1.23	71	289
1.49	82	289.5
1.54	88	292
1.58	87.5	294.5
2.02	84	295.5
2.04	82	295
2.09	81	295.5
2.12	84	296.5
2.15	86	297.5
2.17	86.5	297.5
2.23	85	296
2.27	87	297.5
Normal	85	288

TEST NO. 2.

Test No. 2 was made with the complete calorimeter. The boiler was of the cast-iron sectional type, and was employed mainly to supply steam for heating purposes.

The various sections in this boiler are connected by a header at the bottom, into which the feed-water is supplied, and by means of a drum on the top, from which the steam is carried away into the building. The principal features of the boiler are shown in the appended figure (Fig. 144). The calorimeter was attached to the drum at a point 3½ feet distant from the front end, and the form of the apparatus used and the manner of attachment are shown in the sketch.

At the time of making Test No. 2*a*, the boiler, which was new, had been in use a considerable length of time without blowing off, and, as the test proved, it was discharging a large quantity of wet steam. The water drawn from the drip-box, as will be seen in the last column of Table No. 2*a*, which gives the results of this test, varied from 0.08 of a pound per hour to 4.5 lbs. per hour, and these quantities, referred to the calculated rate at which the

steam passed through the instrument—viz., 45.6 lbs.—show a variation of moisture, as measured by the drip-box alone, of from 0.17 of one per cent., the observation at 3.20, to about 10%, or that given by the observation at 3.32. The lower thermometer in the heat-gauge, as will be seen by the record, showed a nearly constant temperature. In spite of the great variation in the actual percentage of moisture as shown by the drip-box, the temperature changed only from 264° to 269°. The normal readings were not observed, but the indications point to a normal of 272° for a temperature of 304° by the upper thermometer, and to a rate of condensation in the drip-box of 0.08 of a pound per hour, which was the minimum rate shown at 3.20 P.M. The average of all the readings by the lower thermometer is 266.9°, and for the drip-box 1.68 lbs. per hour. The percentage of moisture shown by the heat-gauge here is $272° - 266°.9 = 5.1°$ for the cooling, and this represents $\frac{5.1}{21.5} \times 100 = 0.24$ of one per cent. of moisture. The water drawn from the drip-box, and corrected for the assumed radiation, is $1.68 - 0.08 = 1.6$ lbs. per hour, and this represents $\frac{1.6}{45.6} \times 100 = 3.51\%$ of moisture. The two percentages added together make a total of 3.75%.

DIMENSIONS AND OTHER DATA REGARDING BOILER REFERRED TO IN FIG. 144, TEST NO. 2.

1.	Size of grate	5 ft. 10 in. by 5 ft. 8 in.
2.	Extreme width of each section	45 "
3.	Extreme height of each section	4 ft. 4 in.
4.	Number of sections	32
5.	Approximate area water-heating surface each section	20.26 sq. ft.
6.	Approximate area steam-heating surface each section	7.6 " "
7.	Total heating surface of whole boiler	899 " "
8.	Area of grate surface	17.5 " "
9.	Ratio of total heating surface to grate surface	51.1 to 1
10.	Water evaporated per square foot of heating surface from 100° at 70 lbs. pressure per hour	2.4 lbs.
11.	Kind of coal used	Anthracite broken

Fig. 144.

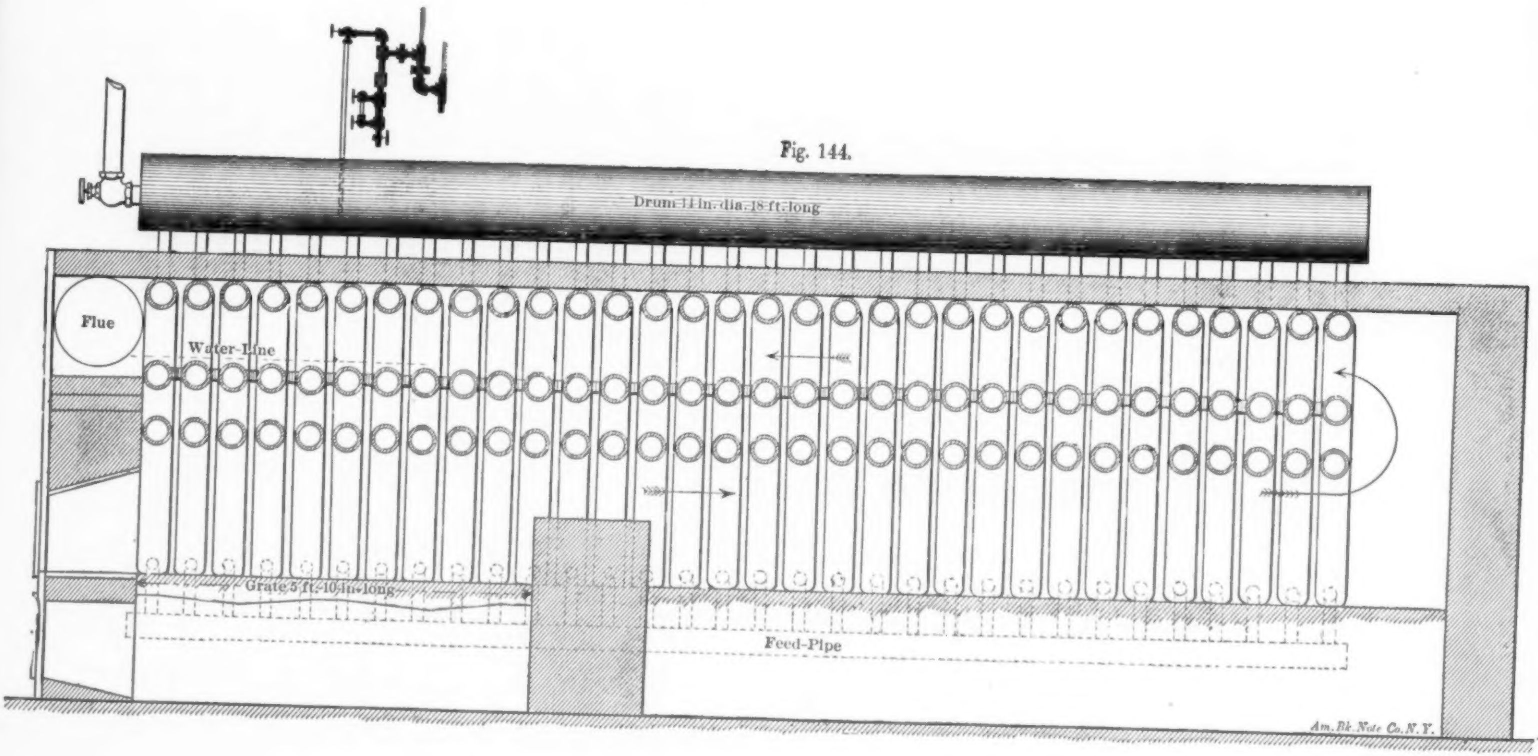

Drum 11 in. dia. 18 ft. long
Flue
Water-Line
Grate 5 ft. 10 in. long
Feed-Pipe
Am. Bk. Note Co. N. Y.

TABLE No. 2*a*.

CAST-IRON SECTIONAL BOILER (Fig. 144) PREVIOUS TO BLOWING OFF AND RENEWING WATER.

Complete Calorimeter in Use.

Time.	Upper Thermometer.	Lower Thermometer.	Weight of Pail containing Water drawn off from Drip-box.	Net Weight of Water drawn from Drip-box during Preceding Interval.	Rate per Hour at which Water is Drawn off from Drip-box, lbs.
2.50	304	264	6 oz.	....	....
2.55	304	266	$8\frac{3}{4}$ "	$2\frac{3}{4}$ oz. in 5 min.	2.03
2.58	303	267	$4\frac{1}{4}$ "	....	....
3.10	300	266	$7\frac{3}{4}$ "	$3\frac{1}{2}$ oz. in 12 min.	1.09
3.15	304	266	$8\frac{1}{2}$ "	$\frac{3}{4}$ " 5 "	0.56
3.20	304	267	$8\frac{5}{8}$ "	$\frac{1}{8}$ " 5 "	0.08
3.27	303	267	12 "	$3\frac{3}{8}$ " 7 "	1.81
3.32	299	266	18 "	3 " 5 "	4.50
3.38	302	265	$6\frac{3}{4}$ "	....	....
3.43	305	268	11 "	$4\frac{3}{4}$ oz. in 5 min.	3.56
3.50	306	268	$13\frac{1}{4}$ "	$2\frac{1}{4}$ " 7 "	1.20
4.02	304	269	17 "	$3\frac{3}{4}$ " 12 "	1.17
4.03	...	...	$6\frac{1}{4}$ "	....	....
4.07	304	269	$7\frac{1}{4}$ "	1 oz. in 4 min.	0.94
4.16	306	267	11 "	$3\frac{3}{4}$ " 9 "	1.56
4.25	303	268	$15\frac{1}{4}$ "	$4\frac{1}{4}$ " 9 "	1.71
Av...	303.4	266.9	......	....	1.68

Normal readings were not observed.

Weight of steam passing through orifice ($\frac{1}{8}$ in. approximate diameter), calculated for 72 lbs. pressure (absolute), $\frac{72}{70} \times 3{,}600 \times .0123 = 45.6$ lbs. per hour.

Test No. 2*b* was made on the same boiler, using the same apparatus, and all the conditions were the same, excepting that during the interval between the two tests the boiler had been blown off and the water renewed. The results of the test, made under these circumstances, are given in Table No. 2*b*.

The average readings of the two thermometers on this test are 298.7° for the upper thermometer, and 264.7° for the lower thermometer. The normal for 298.7° may be taken at 269°, and the cooling below this point is 4.3°, which corresponds to $\frac{4.3}{21.5} = 0.2$ of one per cent. of moisture. The average of the drip-box readings gives 0.23 of a pound per hour. Using the assumed quantity for radiation that was taken before, the net quantity due to moisture in the steam becomes $0.23 - 0.08 = 0.15$ lb., and this represents $\frac{0.15}{45.6} \times 100 = 0.33$ of one per cent. of moisture.

The sum of the two percentages is 0.33 + 0.2 = 0.53 per cent., which represents the total moisture.

TABLE No. 2*b*.

CAST-IRON SECTIONAL BOILER (same as preceding) AFTER BLOWING OFF AND RENEWING WATER.

Complete Calorimeter in Use.

Time.	Upper Thermometer.	Lower Thermometer.	Weight of Pail containing Water drawn off from Drip-box.	Net Weight of Water drawn from Drip-box during Preceding Interval.	Rate per Hour at which Water is Drawn off from Drip-box, lbs.
9.15	300	265	16.5	....	....
9.35	296	263	11.75	1.25 oz. in 20 min.	0.23
9.55	298	264	13	1.25 " 20 "	0.23
10.15	302	266	14.5	1.5 " 20 "	0.28
10.45	296	264	16.25	1.75 " 30 "	0.22
11.15	300	266	17.75	1.5 " 30 "	0.19
Av...	298.7	264.7	.....	....	0.23

TEST NO. 3.

Test No. 3 was made on a water tube boiler of well-known type, the general features of which are shown in Fig. 145. Appended is a table giving the principal dimensions of the boiler and the general conditions under which it was operated. In this test the complete calorimeter was at first in use, and the readings which were taken are as follows:

TABLE No. 3*a*.

WATER TUBE BOILER (Fig. 145).

Time.	Upper Thermometer.	Lower Thermometer.	Water drawn from Drip-box in Five Minutes.	Rate per Hour at which Water is Drawn from Drip-box.
START CALORIMETER AT 9.05.				
9.35	348	280	0.5 oz.	0.36 lb.
9.48	348	286	0.5 "	0.38 "
10.05	349	286	1.0 "	0.75 "
STOP AT 10.06—START AT 12.55.				
1.10	350	287	0.75 lb.	0.55 lb.

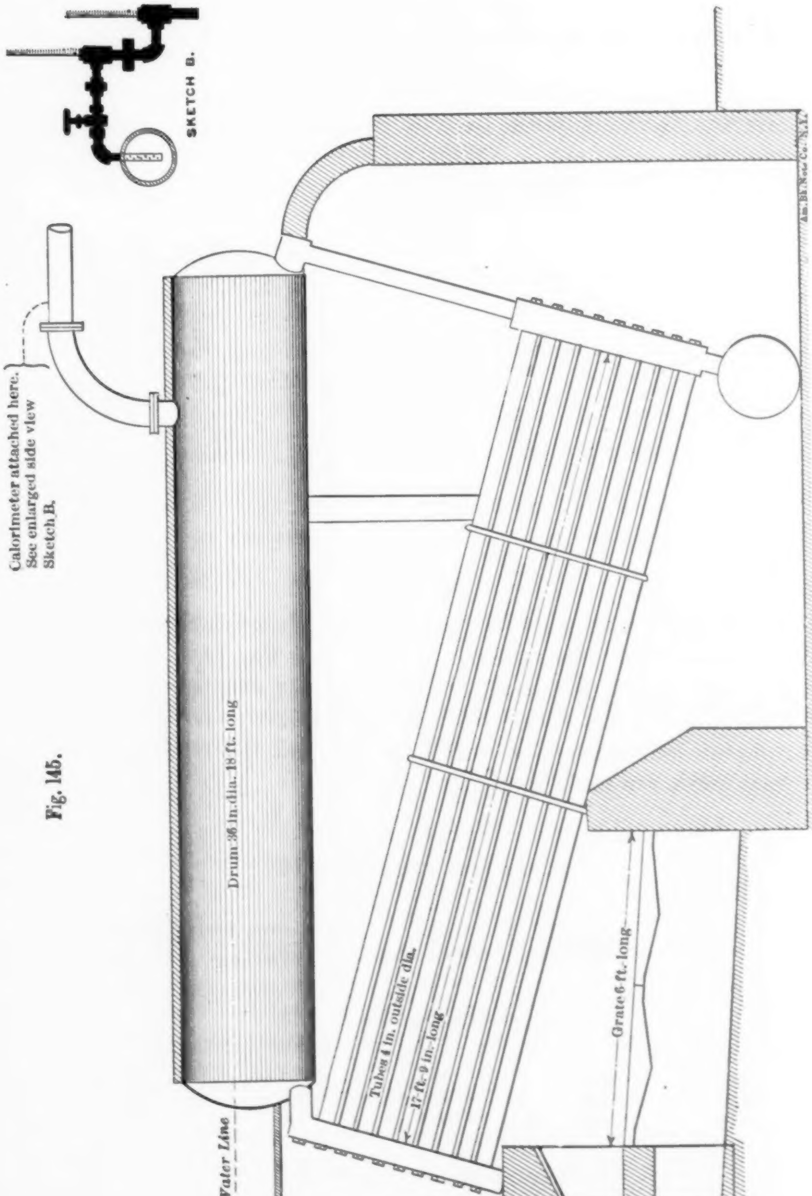

SKETCH B.
Calorimeter attached here.
See enlarged side view
Sketch B.
Drum 36 in. dia. 18 ft. long
Tubes 4 in. outside dia.
17 ft. 9 in. long
Grate 6 ft. long
Water Line

Fig. 145.

There were no normal readings taken when using the complete apparatus, as it was evident that the percentage of moisture was not high enough to call for anything more than the heat-gauge. The drip-box was consequently removed, and the remaining readings, which are given below, were taken, using simply the heat-gauge.

TABLE No. 3*b*.

Time.	Upper Thermometer.	Lower Thermometer.
2.50	346	277
3.08	348	273
3.20	350	278
3.35	350	274
3.40	350	271
3.45	346	277
Average.........	348.7	275
Normal..........	349	289

The average readings of the upper thermometer on this test were 348.7°, and of the lower thermometer 275°, the range of this latter being from 271° to 278°. The normal for the lower thermometer, which was found by taking indications when no steam was being drawn from the boiler, a steady pressure being maintained, was 289°. The average cooling effect, due to moisture in the steam, is $289° - 275° = 14°$; and this, divided by the coefficient for 350°, which is 20.6, gives 0.68 of 1%.

On this test the quantity of steam passing through the orifice, which was $\frac{1}{8}$ of an inch in diameter, was determined by carrying it into a barrel of water resting on scales. The quantity condensed was 14.75 lbs. in 10 minutes, which is at the rate of 68.5 lbs. per hour.

DIMENSIONS AND OTHER DATA REGARDING BOILER REFERRED TO IN TEST NO. 3.

1. Number of tubes, 4 in. outside diameter........................60
2. Length of tubes..16 ft.
3. Diameter of each drum (2 in all)..........................36 in.
4. Length of drum.. 18 ft.
5. Size of grate..................................6 x 4 ft. 10 in.
6. Area of heating surface.................... ....1,251 sq. ft.
7. Area of grate surface..................................29 "

8. Ratio of heating surface to grate surface..................43.1 to 1
9. Kind of coal used............................Pittsburg bituminous
10. Percentage of ashes.......................................6.9%
11. Coal per hour per square foot of grate....................13.1 lbs.
12. Water evaporated per hour per square foot of heating surface from 100° at 70 lbs. pressure...............................2.4 lbs.
13. Average temperature of flue gases.........................540°
14. Water evaporated per pound of combustible from and at 212°......................................9.17 lbs.

TEST NO. 4.

Test No. 4, like the preceding one, was made on a water tube boiler of the style shown in the preceding cut. On this test the heat-gauge only was in use, and the point where it was applied is shown in Fig. 145. Appended is a table showing the general dimensions of the boiler and the conditions under which it was operated.

DIMENSIONS AND OTHER DATA REGARDING BOILER REFERRED TO IN TEST NO. 4.

1. Number of sections..14
2. Number of tubes, 4 in. outside diameter, in each section.........9
3. Total number of tubes..126
4. Diameter of drum..36 in.
5. Size of grate..7 x 8.5 ft.
6. Area of heating surface..............................2,765 sq. ft.
7. Area of grate surface.....................................59.5 "
8. Ratio of heating surface to grate surface.................46.5 to 1
9. Kind of coal used...................George's Creek, Cumberland
10. Percentage of ashes..6.3%
11. Coal consumed per hour per square foot of grate..........17.9 lbs.
12. Water evaporated per square foot of heating surface from 100° at 70 lbs. pressure per hour...........................3.38 lbs.
13. Average temperature of flue gases..........................545°
14. Average draught suction...................................0.49 in.
15. Water per pound of combustible from and at 212°........10.42 lbs.

On this test readings of the instrument were taken at intervals of from one to five minutes during most of a ten-hours' run, and the full set of observations is given in Table No. 4. Remarks are given, in connection with many of the readings, as to the condition of the fire, height of water in the gauge-glass, and other information. It will be seen from this table that the quantity of moisture in the steam was quite variable, though never excessive. The smallest indication of the lower thermometer was that taken at 4.56 P.M., when the reading was 249°, and the highest indi-

cation was at 7.02 A.M., when the reading was 279°. The range corresponds to a little over 1% of moisture. Using the normal of 280° for a temperature of 331° by the upper thermometer, which was found at a time when the boiler was discharging little, if any, steam, the lowest reading, of 249°, gives a cooling effect due to moisture of 280° − 249° = 31°; and the highest reading gives 280° − 279° = 1° for the cooling effect of moisture. The coefficient for 330° is 21°—that is, the cooling due to 1% of moisture. The two extreme percentages of moisture are, therefore, $\frac{31}{21} = 1.48\%$, and $\frac{1}{21} = 0.048\%$.

The variations in the indications of the lower thermometer were so marked, and occasionally so rapid, that an attempt was made to ascertain whether these variations could be accounted for by any differences in the condition of the fire, the height of water, or the manner of feeding the water. At one time it was thought that the lowering of the thermometer was caused by an increased activity of the fire. Notice the reading at 8.42½ A.M., which was 274°, and the next reading, which fell to 251°, 7½ minutes afterward, and between these two readings the fire was shoved back and new coal added. The next time, however, that the fire was shoved back, which was at 9.12 A.M., there was no immediate change in the indication of the thermometer, though at 9.20 the reading had fallen to 259°, and this fall may finally have been due to the increased activity of the fire. A little farther on, at 9.55, the reading was 277°. Shortly afterward the fire was shoved back, and immediately the temperature fell to 257°, and two minutes later to 253°. Take the reading, however, at 12.28, when the fire was treated in the same manner—there was no fall in the temperature, even after slicing the fire, and even when the water was pumped up to quite a high point.

TABLE No. 4.

WATER TUBE BOILER (Fig. 145).

Time.	Upper Thermometer.	Lower Thermometer. Normal 279, with Upper Thermometer 330. Say, 280 at 331.	Remarks as to Height of Water, State of Fire, and other Observations.
6.33	329	277	
7.02	331	279	
7.35	331	271	
8.12	331	255	
8.20	331	272	
8.22½	331	276	
8.30	331	273	
8.32½	331	276	
8.40	331	274	
8.42½	331	274	Shove back and fire at 8.46.
8.50	330	251	
8.52½	330	252	
8.55	331	264	
9.00	329	275	
9.05	330	271	
9.10	330	277	
9.13	330	277	Shove back at 9.12.
9.14	331	275	Firing; water, 3 inches.
9.15	331	276	
9.20	331	259	
9.22½	331	251	Water, 4½.
9.25	331	265	
9.31	331	275	Shove back and fire, 9.32–33.
9.34½	330	275	Water, 5½.
9.35	330	271	
9.37	331	268	
9.40	331	275	
9.45	331	276	
9.50	331	276	Front fired; water, 4 inches.
9.55	331	277	Water, 3 inches.
10.01½	331	257	Shove back and fire, 9.58–10.00.
10.03½	331	253	Water, 4 inches; feeding fast.
10.09	331	264	
10.12	331	275	
10.17	330	277	
10.22	331	271	Front shoved back and fired.
10.28	330	273	Water, 5½.
10.32	331	252	" 4½.
10.35	331	271	
10.41	330	275	
10.45	330	260	Water, 6½.
10.50	331	256	Front shoved back and fired.
10.55	331	250	Water, 6; pump slow.
11.07	330	263	Feeding fast; height, 4 inches.
11.15	331	260	
11.15½	331	251	Height, 5¾.
11.18	330	256	Front fired; height, 6.
11.20½	...	263	
11.22	331	258	

TABLE No. 4.—*Continued.*

Time.	Upper Thermometer.	Lower Thermometer. Normal 279, with Upper Thermometer 330. Say, 280 at 331.	Remarks as to Height of Water, State of Fire, and other Observations.
11.23	331	253	
11.32	331	259	
11.35	331	274	Front fired ; height, 6.
11.38	330	276	
11.41	330	275	
11.43	331	256	Water, 7½.
11.45	331	261	Front shoved back.
11.47	331	267	
11.50	331	260	Height, 6½.
11.55	331	262	" 6½.
12.00	331	253	" 7.
12.06	332	277	" 7 ; damper shut.
12.10	332	259	" 7 " open.
12.15	332	260	" 6½.
12.22	332	252	" 5.
12.28	331	275	Front shoved back and firing.
12.31	330	277	Height, 5.
12.34	329	275	
12.38	328	275	Front sliced ; height, 7.
12.40	327	271	
12.45	327	273	
12.51	329	275	
12.59	330	258	Height, 7.
1.13	330	275	" 7.
1.21	330	276	" 8.
1.30	330	277	" 4.
1.38	330	261	Feeding fast ; height, 5.
1.43	329	262	Front shoved back and fired.
1.50	330	264	Feeding fast ; height, 6.
1.57	330	256	Height, 5.5.
2.08	331	266	" 5.5.
2.14	331	253	" 5.2.
2.20	330	273	Shoved back, 2.21.
2.22	329	268	Fired, 2.22½.
2.23	329	261	
2.24	329	259	
2.25	329	252	
2.26½	329	252	Height, 6.5.
2.28	329	255	Slow down pump, 2.29½.
2.30	330	269	
2.35	331	276	
2.46	331	261	Height, 3.
2.52	331	263	Shove back, 2.50.
2.54	330	256	
3.04	331	266	Height, 3.
3.09	330	276	" 3.
3.20	331	255	" 2.5.
3.25	331	255	" 3.
3.32	331	273	" 4.
3.40	331	275	Shoved back and fired, 3.35 ; height, 6.5.
3.45	331	277	Height, 5.

TABLE No. 4.—*Concluded.*

Time.	Upper Thermometer.	Lower Thermometer. Normal 279, with Upper Thermometer 330. Say, 280 at 331.	Remarks as to Height of Water, State of Fire, and other Observations.
3.53	331	268	Height, 3.5.
3.59	331	270	" 4.5.
4.06	331	264	" 3.5.
4.11	331	271	Shoved back, 4.10; fired, 4.11½.
4.12	331	270	Height, 3.5. (?)
4.13	331	271	
4.14	331	266	Height, 5.5.
4.15	331	262	" 6.
4.17	330	273	
4.24	331	265	Height, 4.5.
4.28	330	273	" 4.
4.39	326	275	" 4.
4.45	326	275	" 4.5.
4.53	330	258	Front shoved back and fired, 5.30.
4.56	330	249	Height, 6.5.
5.04	330	276	" 5.5.
5.16	331	276	
5.22	331	275	Height, 5.5.
5.28	333	276	Damper shut.
5.33	331	278	Height, 5.4.

TEST NO. 5.

Test No. 5 was made on a plant of four horizontal return tubular boilers, the plan and longitudinal cross-section of which are shown in Fig. 146 and Fig. 147. The whole plant consisted of six boilers, but only four of them were in use, and these were employed solely for supplying a 22″ and 44″ x 60″ compound engine, developing an average of 618 I.H.P. The ratio of grate surface to heating surface was 44.2 to 1; the rate of combustion was 10 lbs. of Cumberland coal per square foot of grate surface per hour; the percentage of ashes was 8.7; the evaporation from 100° at 70 lbs. pressure per square foot of heating surface per hour was 1.97 lbs.

On this test the heat-gauge alone was in use for two periods of 15 minutes each, and the two thermometers showed temperatures which were practically constant, viz., 341° and 283° respectively The normal reading of the lower thermometer was 287°. From these figures the cooling effect of the moisture was

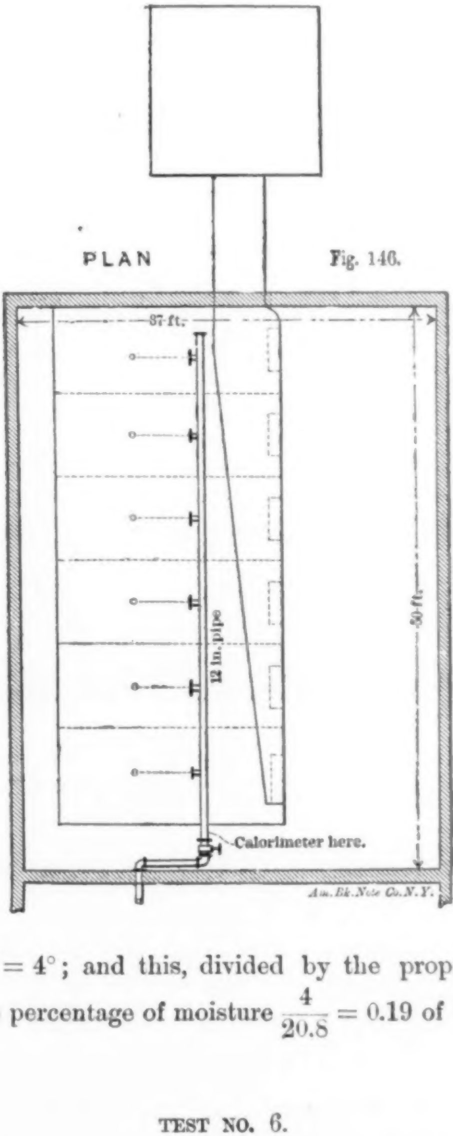


Fig. 146.

$287° - 283° = 4°$; and this, divided by the proper coefficient, gives for the percentage of moisture $\frac{4}{20.8} = 0.19$ of one per cent.

TEST NO. 6.

Test No. 6 was made on a plant of eleven horizontal return tubular boilers, which were used mainly for supplying steam to a 700 H.P. condensing engine. The general location of the boilers and the arrangement of the steam-piping are shown in Fig.

148. Two of the boilers were provided with a steam dome, and they all discharged the steam into a drum, which, however, was not very well drained on the end where the steam was taken off. On this test simply the heat-gauge was used, the manner of attachment being shown in the sketch. The full set of observa-

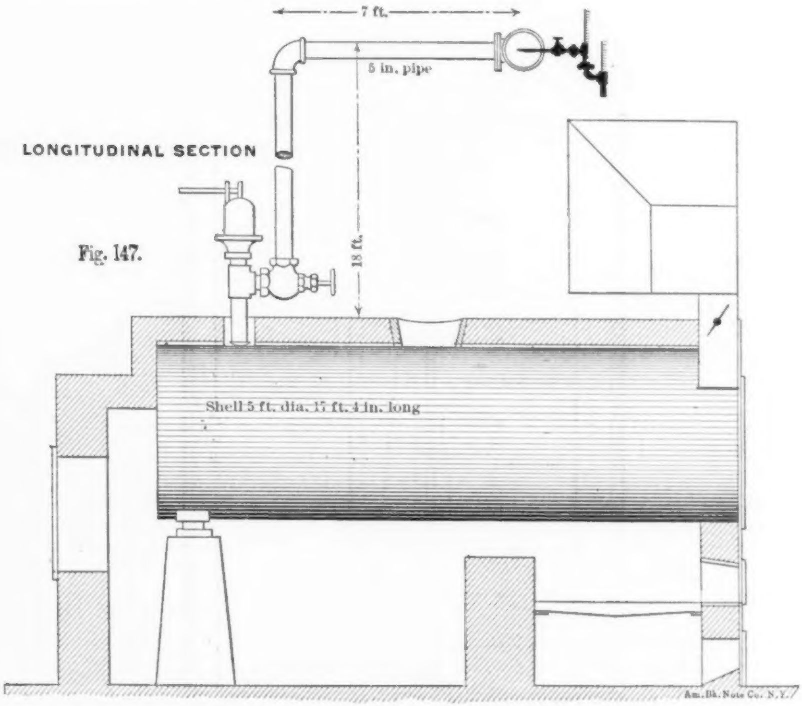


Fig. 147.

tions is given in Table No. 5. The average of all the readings is 312° for the upper thermometer and 243° for the lower thermometer. The normal at 312° may be taken at 268°. The cooling effect of the moisture is, therefore, 268° − 243° = 25°, which corresponds to $\frac{25}{21.3} = 1.17\%$. The extreme range of the indications of the lower thermometer are from 226° for the minimum to 252° for the maximum.

Fig. 148.

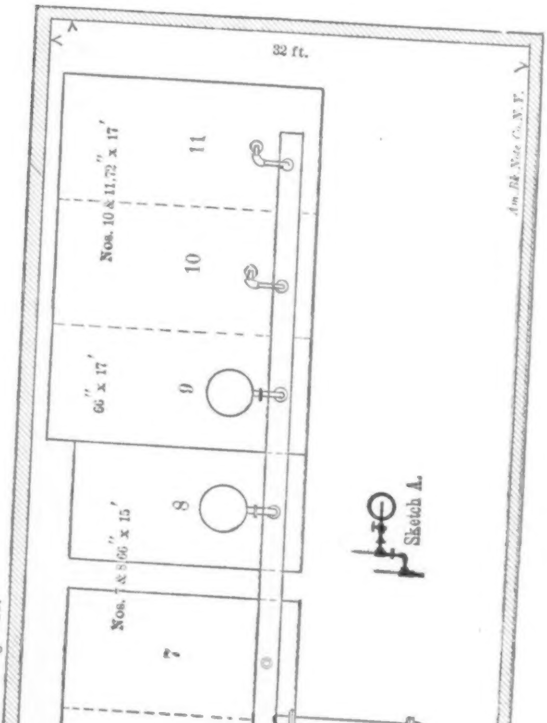

32 ft.
57 ft.
Nos. 10 & 11,72" x 17'
66" x 17'
Nos. 7 & 8 66" x 15'
Nos. 3-6,54" x 15'
Nos. 1 & 2,48" x 15'
1
2
3
4
5
6
7
8
9
10
11
Sketch A.
10" pipe
Calorimeter here
See Sketch A.

TABLE No. 5.

PLANT OF ELEVEN HORIZONTAL RETURN TUBULAR BOILERS (Fig. 148).

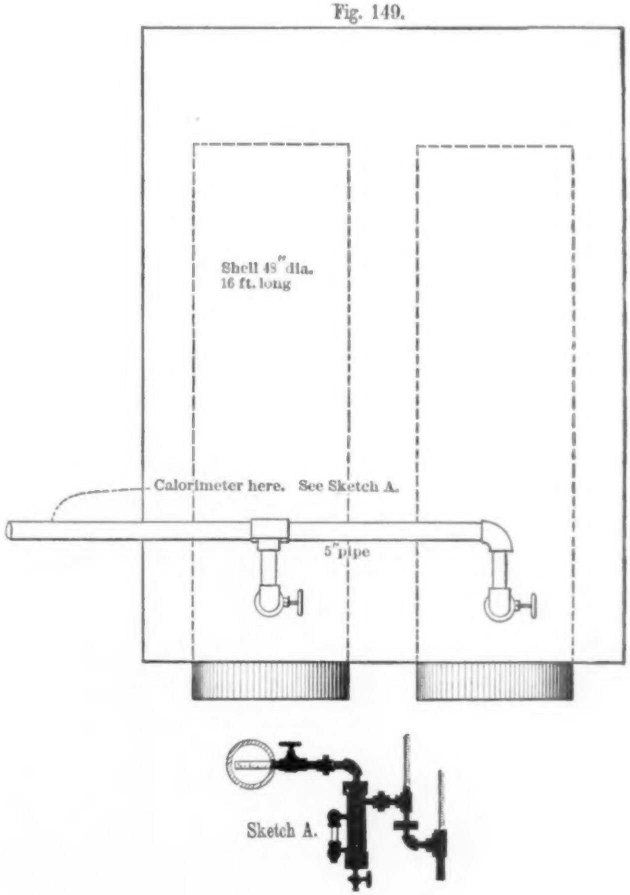
Time.	Upper Thermometer.	Lower Thermometer.
9.55	312	235
10.00	312	226
10.05	313	240
10.10	312	245
10.12½	312	240
10.15	313	241
10.17½	312	241
10.20	312	242
10.22½	313	241
10.25	313	240
10.27½	312	241
10.30	312	238
10.32½	313	239
10.35	313	245
10.37½	312	248
10.40	312	249
10.42½*	311	248
10.45	314	247
10.47½	315	250
10.50	314	252
10.52½	314	249
10.55	314	249
10.57½	314	249
11.00	313	248
11.02½	314	246
11.05	314	246
11.07½	309	243
11.10	312	239
11.12½	313	238
11.15	313	239
11.17½	312	240
11.20	311.5	238
11.22½	312	233
11.25	312	229
11.27½	313	234
11.30	313	243
11.32½	313	244
11.35	312	245
11.37½	313	247
11.40	312	247
11.42½	313	247
11.45	312	245
11.47½	312	243
11.50	312	242
11.52½	311	243
11.55	311	248
Normal	315	271
Average............	312	243

* Between 10.42½ and 11.05 thermometers interchanged.

TEST NO. 7.

Test No. 7 was made on a plant of two horizontal return tubular boilers, and in this case the complete form of the instrument was used. The plan of the boilers and the arrangement of the

Fig. 149.



steam-piping, together with the location of the calorimeter and the form of instrument used, are shown in Fig. 149. The full set of observations is given in Table No. 6. Considerable interest attaches to this test on account of the priming of one of the boilers, which was brought about by design. This boiler was filled with water to nearly the top of the glass, and the bituminous coal

fire was barred up, the damper opened wide, and the boiler made to do its maximum amount of work. The time when this occurred was 2.05 P.M., and shortly afterward the quantity of water collecting in the drip-box began to increase, and for the seven and a half minutes between 2.15 and 2.22½, 18.9 ounces of water were withdrawn. This represents 9.28 lbs. per hour, or about 18.5% of moisture, the quantity of steam used being estimated at 50 lbs. per hour.

Taking the ordinary indications of the instrument during the 20 minutes' time between 12.55 and 1.15 P.M., the drip-box discharged 2.7 ounces, or 0.51 of a pound per hour. The average reading of the upper thermometer was 304.9° and of the lower thermometer 268.2°. Comparing these observations with the normal readings, it is seen that there was but a trifling indication of moisture.

It may be added that the priming was found to be due, when the water was carried too high, to the presence of vegetable matter in the water and to too infrequent blowing off.

TABLE No. 6.

TWO HORIZONTAL RETURN TUBULAR BOILERS (see Fig. 149). COMPLETE CALORIMETER IN USE.

Time.	Upper Thermometer.	Lower Thermometer.	Height of Water in Glass, in Sixteenths of an Inch.	Remarks.
12.22½	306	265	3	Practically no steam drawn off from boilers between 12.25½ and 12.47½.
12.25	304.5	267	5	
12.27½	302	267	7 scant	
12.30	301	267	8 scant	
12.32½	303	267	10	
12.35	305	267	13 +	
12.37½	303	268	15 +	
12.40	304	268.5	17 scant	
12.42½	304	268.5	18 +	
12.45	303	268	20	
12.47½	302	267	2 +	Draw off from drip-box at 12.46, 2.7 ounces.
Average normal readings	303.6	267.3	0.39 lb. per hour	
12.50	302	267	4 +	
12.52½	302	267	6 scant	
12.55	303	267	8 +	Engine started at 12.55.
12.57½	303	267	11	
1.00	303	267	14	
1.02½	305	268	16 +	

TABLE No. 6.—*Concluded.*

Time.	Upper Thermometer.	Lower Thermometer.	Height of Water in Glass, in Sixteenths of an Inch.	Remarks.
1.05	306	268	19	
1.07½	307	269	21 +	
1.10	307	269	24	
1.12½	306	270	7	Draw off from drip-box, at 1.11, 2.7 ounces.
1.15	304	269	8 +	
1.17½	302	268	9 +	
1.20	301	268	10 +	
1.22½	302	267	12 +	
1.25	302	267	14 +	
1.27½	302	267	16 +	
1.30	304	267	19	
1.32½	304	268	22	
1.35	304	268	23	
1.37½	303	268	5.5	Draw off from drip-box, at 1.36, 2.7 ounces.
1.40	301	268	6 +	
1.42½	300	267	8	
1.45	298	266	10	
1.47½	298	266	10 +	
1.50	300	265	13	Water being pumped to a high point.
1.52½	302	265	16	
1.55	304	266	19	
1.57½	306	267	22	
2.00	306.5	268	24	Draw off from drip-box, at 2.01, 2.7 ounces.
2.02½	307	269	8	
2.05	307	270	11	At 2.05 damper of one boiler shut. Fire in the other boiler barred up and water at a high point.
2.07½	308	270	22 +	
2.10	306	270	29	
2.12½	308	270	12	
2.15	307	270	14	
2.22½	306	255	12	Draw off from drip-box, at 2.11, 2.7 ounces.
2.25	308	266	15	
2.27½	310	270	18 +	Between 2.15 and 2.22½ draw off from drip-box 18.9 ounces.
2.30	310	270	20 +	
2.32½	310	270	21 +	
2.35	309	271	23	At 2.25 height of water in the active boiler had fallen 4 inches.
2.37½	310	271	7	
2.42½	301	268	8	
2.45	298	267	8 +	Draw off from drip-box, at 2.36, 2.7 ounces. At 2.42½ dampers of both boilers open.
2.47½	299	266	10	
2.50	301	266	12	
2.52½	305	267	17	
2.55	308	266	21	
2.57½	310	269	23	
3.00	312	270	24 +	

TEST NO. 8.

Determination of the Coefficient and Test of the Accuracy of the Heat-gauge.

The series of experiments which the author has made to determine the proper coefficient to be used in estimating the percentage of moisture from the readings of the thermometers in the

heat-gauge have not, as already stated, been made sufficiently complete thoroughly to settle the quantities for all conditions; but, so far as they have been completed, they are believed to be reliable, and applicable to this form of instrument, within the range of operations chosen. The idea which has been worked out was to conduct the steam to the calorimeter through a certain amount of piping which, by the loss of heat due to radiation, would produce a certain constant rate of moisture. This was first drawn off from the drip-box, and its quantity determined. It was then conducted into the heat-gauge, and the effect of this upon the thermometers was observed.

The apparatus used is shown in Fig. 150. The drip-box is represented at *E* and the heat-gauge at *F*, and the two valves *G* and *H* were provided, the former to draw off the moisture from the drip-box, and the latter to carry the moisture into the heat-gauge. Following back to the right-hand end of the figure, the steam is taken first from a horizontal 2-inch pipe, which was supplying, during most of the experiments, an engine, and during the remaining ones it carried off a sufficient quantity of steam to produce a proper circulation. It dropped vertically through a $\frac{1}{2}$-inch pipe into a well, from the bottom of which an open pet-cock discharged any water of condensation carried along with the steam. This well was made of $1\frac{1}{2}$-inch pipe; from the top of the well a 1-inch pipe, 8 feet long, inclined so as to drop 14 inches, carried the steam to a $\frac{1}{2}$-inch pipe at the left, which, in turn, supplied the top of the drip-box; the bottom of the drip-box was provided with the glass attachment shown, which is a slightly different arrangement from that shown in the drawing at the opening of the paper, although it answers the same purpose. The lower end was connected, through a series of tees, valves, and unions, to the entrance of the heat-gauge *F*. From the top of the drip-box a $\frac{1}{2}$-inch pipe was carried to the heat-gauge, being joined at an intermediate point between the valve *H* and the upper thermometer. The drip-box proper was made of $1\frac{1}{4}$-inch pipe, and the various sizes of fittings used and the dimensions are indicated in the sketch. On leaving the heat-gauge the steam was conducted through a rubber tube into a coil of iron pipe placed in a barrel, and the outlet of the coil discharged through the side of the barrel to the atmosphere. A current of cold water was passed through the barrel, around the coil, and emptied to waste. Attached to the vent of the drip-box was a coil of $\frac{1}{4}$-inch pipe,

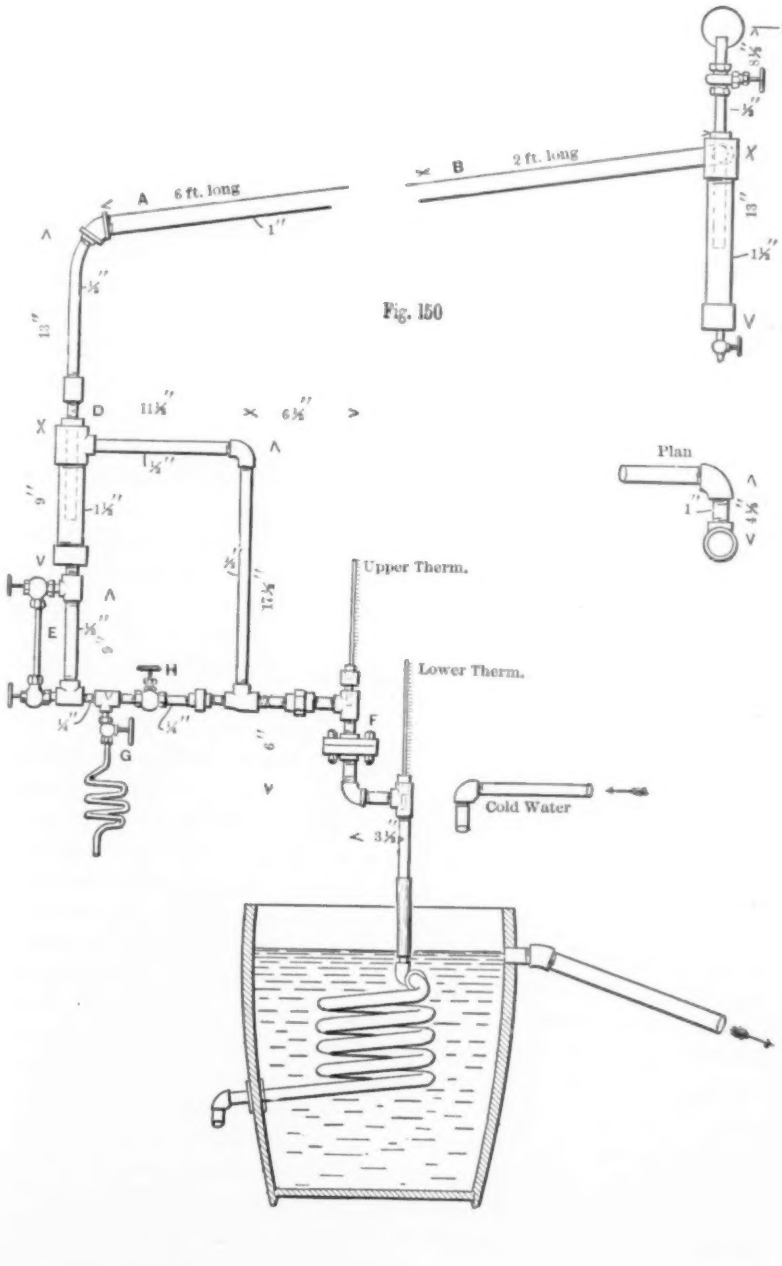

A
6 ft. long
1"
B
2 ft. long
2"
X
13"
1½"
V
13"
D
11½"
6½"
X
Plan
1"
4½"
9"
1½"
V
E
H
17½"
Upper Therm.
Lower Therm.
F
G
6"
Cold Water
3½"

Fig. 150

which was covered with wet cotton-waste, the object of which was to keep the heated water which was discharged through the valve from re-evaporating. The apparatus was all thoroughly covered with hair felting, and on one series of tests the whole of the pipe *A B* was covered, while on another series the part marked *A* was covered and the part marked *B* was bare. The felting was three-quarters of an inch thick.

The method of carrying on the test was to first allow the moisture which entered the drip-box to escape from the valve *G* directly into the atmosphere, the valve *H* being shut. The level of the water was maintained at a fixed point in the glass; readings of the two thermometers were taken every two or two and a half minutes, and the quantity of water drawn off at *G*, as also that which was discharged through the condensing coil, was measured every five or six minutes. The test was usually continued for about one hour's time; some of the tests, however, being of 30 minutes' duration. The first test having been completed, the valve *G* was shut and the valve *H* opened, and the test was continued with no other change of conditions. The opening of the valve *H* was regulated so as to keep the height of water in the glass at the same point as on the preceding test.

The average results of the various tests which have thus far been made are given in the appended Table No. 7, following which is a table giving the full series of observations for a single set of tests made on March 9th.

The method of computing the specific heat from the results of the experiments is as follows: Suppose, for the sake of clearness, that the steam has a temperature of 312° by the upper thermometer, the total heat of which, if dry, is 1,209.1 thermal units; suppose, also, that the normal by the lower thermometer for this temperature, for absolutely dry steam passing by the upper thermometer, is 283°. Let C represent the actual proportion of moisture entering the heat-gauge, and k the specific heat of the superheated steam which is to be found. We have now, for the total heat of the steam entering the heat-gauge, $1,209.1 - (C \times 894.2)$, the quantity 894.2 being the latent heat of steam of this temperature. Suppose now, for simplicity, that the steam issuing from the heat-gauge is at a pressure of the atmosphere, and that T represents the actual temperature of the issuing steam, as shown by the lower thermometer; then the total heat of this issuing steam is given by the expression, $1,178.5 + k\,(T - 212)$. Assuming

TABLE No. 7.

TESTS OF COEFFICIENT.

Where Moisture is Discharged.	Condition as to Covering of Supply Pipe.	Date.	Duration.	Temperature by Upper Thermometer, deg.	Temperature by Lower Thermometer, deg.	Weight of Steam, including Moisture, passing through Orifice per Hour, lbs.	Weight of Water drawn from Drip-box per Hour, lbs.	Specific Heat as Applied to the Instrument, calculated from Foregoing Data, lbs.
Through drip-valve.	Wholly covered.	Dec. 12.	1 hour.	312.4	265.4	53.87	0.644	.....
" heat-gauge.	" "	" 12.	1 "	312.2	239.6	53.89	.....	0.415
" drip-valve.	" "	Feb. 21.	2 hours.	312.5	267.0	53.58	0.684	.....
" heat-gauge.	" "	" 21.	2 "	313.8	241.4	54.97	.....	0.414
" drip-valve.	Partly "	" 28.	1 hour.	313.6	266.4	53.94	1.284	.....
" heat-gauge.	" "	" 28.	1 "	313.1	225.0	54.40	.....	0.514
" drip-valve.	" "	Mar. 9.	30 min.	314.1	267.2	54.50	1.350	.....
" heat-gauge.	" "	" 9.	30 "	314.1	223.9	55.32	.....	0.511
" drip-valve.	" "	" 9.	1 hour.	314.3	267.9	55.00	1.309	.....
" heat-gauge.	" "	" 9.	1 "	314.3	227.1	55.82	.....	0.519

no loss by radiation, these two quantities are equal; in other words, $1{,}209.1 - (C \times 894.2) = 1{,}178.5 + k\,(T - 212)$. We may substitute for 1,209.1 its equivalent—that is, $1{,}178.5 + k\,(283 - 212)$—and we shall then have the following equation: $1{,}178.5 + k\,(283 - 212) - (C \times 894.2) = 1{,}178.5 + k\,(T - 212)$. Reducing and transposing, $k\,(283 - T) = C \times 894.2$; from which we have the equation, $k = \frac{C \times 894.2}{283 - T}$. This equation is true for any other percentage of moisture, c, and its corresponding temperature by the lower thermometer, t; so that, to apply the equation to the exact case under consideration, the specific heat would be found from the final equation, $k = \frac{(C - c)\,894.2}{t - T}$. The quantity in the parentheses, $C - c$, is the proportion of moisture corresponding to the water, which on one experiment is drawn off from the drip-box and on the other experiment carried through the heat-gauge. To show the method of computation, take the first set of experiments made December 12th: The first experiment shows that the quantity of water drawn off from the drip-box is 0.644 lb. per hour. It is presumed that this quantity represents the moisture going into the heat-gauge on the second test, when the total quantity of steam and moisture passing into the heat-gauge, as shown by the condensing coil, was 53.89 lbs. per hour. The proportion of moisture is $\frac{0.644}{53.89} = 0.0119$; this, multiplied by 894, which is the latent heat corresponding to 312.2° by the upper thermometer, gives 10.64. The difference in the readings of the lower thermometers on the two tests is 265.4° − 239.6° = 25.8°. The specific heat of the steam, then, is $\frac{10.64}{25.8} = 0.41$.

TABLE No. 8*a*.

Test of March 9, 1890.

MOISTURE PASSING THROUGH HEAT-GAUGE.

Time.	Upper Thermometer, deg.	Lower Thermometer, deg.	Condensed Steam passing through Orifice, drawn from Condenser, lbs.
11.02	315	223	......
11.04	315	225	......
11.06	314.5	223	......
11.08	314	224.5	5-9.2
11.10	314	224	......
11.12	314	224	......
11.14	313.5	225	5-8.25
11.16	313.5	222	......
11.18	313.5	224.5	......
11.20	314	222	5-8
11.22	314	223	......
11.24	314	225	......
11.26	314	224	5-8.6
11.28	314	224	......
11.30	314	224.5	......
11.32	314	224	5-8.5
Average ..	314.1	223.9	......

TABLE No. 8*b*.

Test of March 9, 1890.

MOISTURE PASSING THROUGH DRIP-VALVE.

Time.	Upper Thermometer, deg.	Lower Thermometer, deg.	Condensed Steam passing through Orifice, drawn from Condenser, lbs.	Weight of Bucket containing Water drawn off from Drip-box, lbs.
12.02	314.5	267	......	16.55
12.04	314+	267+	......	.....
12.06	314+	267+	......	.....
12.08	314+	267+	5-8.5	18.75, low.
12.10	314	267+	......	.....
12.12	313.5	267	......	.....
12.14	314	267	5-5.8	20.8, high.
12.16	314.5	266.5	......	.....
12.18	314.5	267	......	.....
12.20	314.5	267	5-8.75	23.1
12.22	314.5	267+	......	.....
12.24	314	267+	......	.....
12.26	314	267+	5-7	25.25
12.28	314	267+	......	.....
12.30	314	267+	......	.....
12.32	313.5	267	5-6	27.35
Average ..	314.1	267.2	......	.....

It may be added here that the quantities of specific heat given in the table are corrected for the slight differences in some of the readings of the upper thermometer on the two tests of a series.

DISCUSSION.

Mr. D. W. Robb.—I have used the Barrus calorimeter quite extensively in testing stationary boilers for about two years, and although I have not tested its accuracy, except in the way of testing the thermometers, and such tests as can be made by the instrument itself, such as testing the boiler while not doing work—from such tests and other facts which I will refer to, I believe it to give correct indications of the moisture in the steam when careful and frequent readings are taken. In one test which I made, a boiler which gave practically dry steam under ordinary conditions lifted water when a steam-hammer was started, and I was surprised to find how quickly the calorimeter responded to the wet steam, the lower thermometer running down almost instantly from the normal indication due to dry steam to sometimes about 212°. This might indicate a possible error in the calorimeter, under certain circumstances, when the wire-drawing portion of the instrument alone is used, such conditions as a locomotive or other boiler presents under sudden fluctuations of work, and suggested to my mind the use of thin copper or some other metal which would absorb and give off heat readily for the parts of the calorimeter. But so far as my knowledge and experience go, I do not believe there is any instrument or method for determining the quality of steam which will work as accurately as a calorimeter constructed on this principle, if frequent and careful readings are taken.

Mr. D. L. Barnes.—I would like to ask Mr. Barrus his experience with that calorimeter on locomotives.

Mr. Barrus.—I have used the calorimeter, or what I call the heat-gauge part of the instrument, on some locomotive tests which I have been making during the past month. I attached the instrument to the steam-pipe just before it enters the cylinder on one side, and I found that on two different locomotives, on a continuous test of some three hours' duration, readings being taken every two or three minutes, the indication of the lower thermometer never went down below 250°, and that is a pretty certain indication that the percentage of moisture, if there was any there, was very small.

Prof. H. W. Spangler.—I think the mechanical engineers who make tests of boilers are considerably indebted to Mr. Barrus for the long and conscientious work which he has done on this subject of the calorimeter. A few years ago a barrel, a piece of hose, and a thermometer were all which was thought necessary to do all this work. But one who carries on a continuous set of experiments with the old apparatus, even after he reaches considerable skill in the use of it, gets some very unsatisfactory results. So, of course, he hunts around to find what is the next best thing. For a number of years we have had various styles of calorimeters. If we look through our Transactions we can see very clearly the course of work which Mr. Barrus has done in perfecting the calorimeter to its present condition. The last form which we had from him was, I think, an entirely satisfactory one, where it could be applied. This one seems to me to fill the bill completely. Determining the moisture in steam is one of the very important things which we should get as near right as we can, and I think Mr. Barrus has put into our hands an instrument which we can use with comparatively little trouble, and from which we can get very satisfactory results.

Mr. Barrus.—I do not think there is anything more to be added. I would like to thank Professor Spangler for what he has said in such complimentary terms, and all I have to say is that the instrument is very handy and simple. It is easy to be applied, and I think, as its name implies, it is an instrument which can be used universally.

CCCLXXXVIII.

MEMORANDA REGARDING THE INDICATING OF THE ENGINE OF THE STEAMER "CITY OF RICHMOND."

BY GEO. H. BARRUS, BOSTON, MASS.
(Member of the Society.)

THROUGH the solicitation of Mr. Ernest N. Wright, of New York, Mr. Hennessey, General Manager of the Inman & International Steamship Co. (Ltd.), gave the writer a letter of introduction to Mr. George Clarke, Assistant Superintending Engineer of the Inman & International Steamship Co. at Jersey City, and this latter gentleman gave every facility for the desired work. The tests were made during the voyage of the engineers in June, 1889, between New York and Liverpool.

Though it was a somewhat novel experience to me to indicate a marine engine at sea, there is not much to be said as to the manipulation of the indicators which will be of general interest. As a

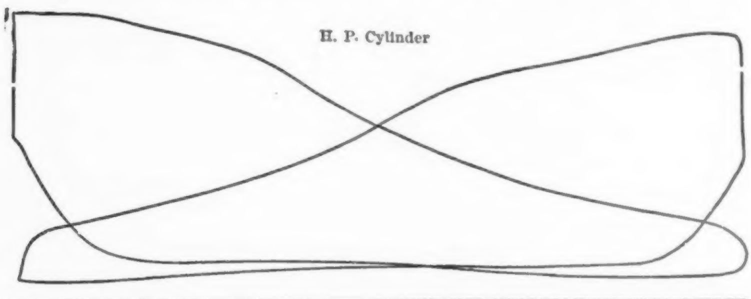


FIG. 151.

matter of record, however, it may be interesting to give copies of the diagrams which were obtained, and some brief statements as to the horse-power, consumption of coal, etc., of the steamer's engines. It ought to be said, at the outset, that the *City of Richmond's* motive power had long been in service, and the conditions were not such as to secure the results which at the present time, on a better class of engines, are generally obtained.

The engine is a compound vertical engine, having one high-pressure cylinder 68 inches in diameter, and one low-pressure cylinder $120\frac{1}{4}$ inches in diameter, both having a stroke of 5 feet. The cylinders are steam-jacketed, but the jackets were out of use. A sample of the diagrams taken is given in Figs. 151 and 152, the scale of the high-pressure cards being 40 and of the low-pressure cards 12.

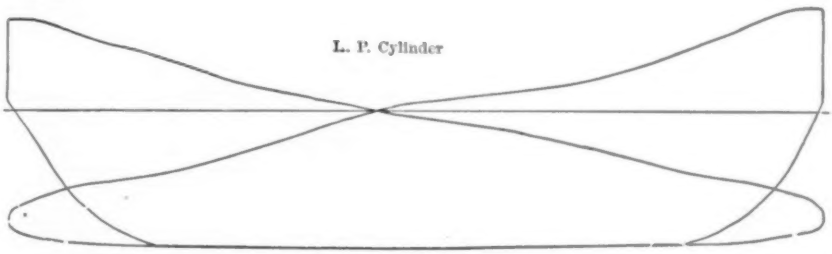


FIG. 152.

The following table gives some of the data regarding the performance:

1. Boiler pressure	78	lbs.
2. Temperature of sea water	70	deg.
3. Temperature of hot well	120	"
4. Temperature of feed water	134	"
5. Vacuum gauge	24	ins.
6. Mean effective pressure from diagrams, H. P. Cylinder	30.31	M. E. P.
7. Mean effective pressure from diagrams, L. P. Cylinder	8.06	"
8. Revolutions per minute	53	rev.
9. Indicated horse-power developed by H. P. Cylinder	1,762.1	I. H. P.
10. Indicated horse-power developed by L. P. Cylinder	1,468.6	"
11. Indicated horse-power developed by both Cylinders	3,230.7	"

The consumption of coal given me was 90 to 100 tons in 24 hours. Taking the smaller figure, the average consumption per hour per indicated horse-power would be about 2.6 lbs.

In a letter received by the writer from Mr. Clarke since the above was put in print, attention is called to the fact that this engine was built by Messrs. Todd & McGregor, of Glasgow, Scotland, and it is one of the first large compound engines ever constructed. Mr. Clarke states that the data here given stand at a disadvantage compared with the records of October, 1888, when the engine developed 4,198 H.P., at a speed of 56 revolutions per minute; the consumption of coal at this time being 112 tons per day, or about 2.4 lbs. per I.H.P. per hour.

CCLXXXIX.

TEST OF A REFRIGERATING PLANT.

BY DE VOLSON WOOD, HOBOKEN, N. J.
(Member of the Society.)

THE substance of the following paper was included in the discussion of one read by Professor Denton at the Erie meeting of this Society, on *The Performance of a 35-Ton Refrigerating Plant*, but the prepared discussion was lost in the mail.

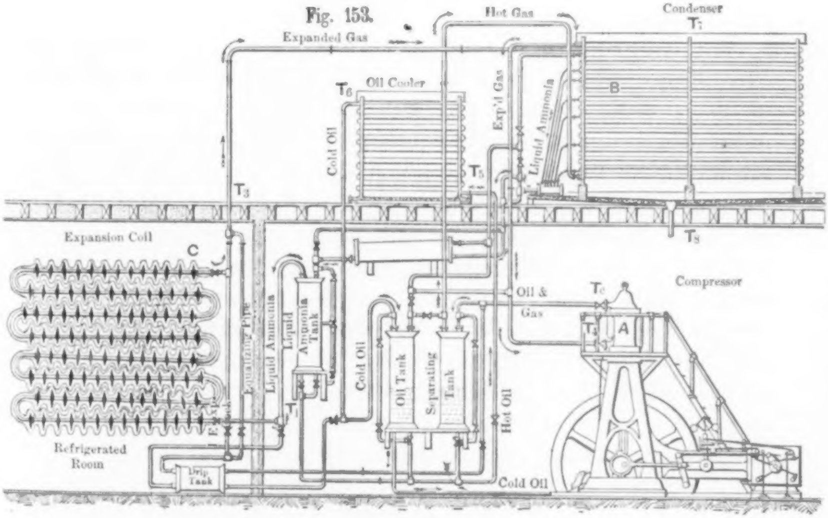
The case was of especial interest to the writer, as it furnished the only data with which I am acquainted for the application of the formulas which were developed by myself in the paper on *Some Properties of Ammonia*, published in Vol. X., pp. 627–647, of the *Transactions* of this Society.

The test was made by Robert M. Anderson and C. H. Page, Jr., upon a nominal 110-ton De La Vergne refrigerating plant in Ballantine's brewery, Newark, N. J., in April, 1887, while they were senior students in Stevens Institute. At that time they had no reliable formulas for reducing that part of the data involving the properties of ammonia. Some of the results deduced by the use of my formulas are given in the third edition of my *Thermo-Dynamics*, but the process by which they were obtained is not explained, for it was supposed that they would appear in the discussion above referred to. The test was conducted with all the care and exactness possible under the peculiar circumstances under which it was made. The thermometers, gauges, and indicators were standardized at the institute.

The plant consisted of two boilers, an ammonia compressor consisting of two single-acting upright cylinders driven by a double-acting horizontal Corliss engine, fresh-water pumps, hoists, and electric dynamos; and in order that the test should involve only the refrigerating plant, it was made on a day when all the other machinery was not used, and only one of the boilers was fired. No brine was used, but the liquid ammonia and the ammonia vapor were circulated through the cooled cellars. The gen-

eral arrangement of the parts is shown in elevation in Fig. 153. A quantity of oil is pumped into each of the compressor cylinders each stroke in sufficient quantity to more than fill the clearance spaces, a part of which passed out of the compressor with the ammonia, and afterward separated from the former in a "separator," and the remainder leaked through the stuffing and ran down the piston-rod. The volume of ammonia per stroke was considered as equal to the piston displacement less the volume of oil forced into the cylinder, although, on account of the leaking of the oil, the volume of ammonia somewhat exceeded this amount.

Fig. 153.



The temperature of the ammonia gas in the pipes was determined by inserting the bulb of a thermometer in a pouch filled with mercury, which latter was bound against the pipe on the outside. The correction to be made to the readings was determined by an experiment in the laboratory, in which the thermometer was placed as nearly as possible in the same condition as in the test, the heated fluid passing through the pipe being water whose temperature was measured directly. The maximum difference between the readings of the outside and inside thermometer was about 2° Fahr. Professor Mayer, according to experiments since made at the institute, gives it as his opinion that if the mercury had been in contact with the pipe, and the pipe well covered by felt for a foot or two each side of the place of the thermometer,

the reading of the thermometer would have been the temperature of the ammonia in the pipe—in which case no correction would have been needed. The plant when tested was in its ordinary every-day working condition, and there was no "coaching" from the manufacturers or others.

WEIGHT OF AMMONIA.

In order to determine the weight of ammonia passing through the compressors, it will be necessary to determine the probable mean temperature of the gas when the cylinder is full and the valve closed.

The temperature of the gas entering the compressor was 57.7° Fahr., mean value, and it was compressed to a temperature of 116.1° Fahr., mean value. When the piston is at the upper end of the stroke the cylinder below it will be full of the gas, which entered at 57.7° Fahr.; but the upper end of the cylinder, being hotter than the lower, and being about 116° Fahr., the temperature of the upper portion of the gas will exceed 58° Fahr., and may be nearly 116° Fahr. The piston also being heated will, as it descends, impart heat to the gas, the piston becoming cooler in its descent and the gas at the lower end being heated. If the temperature of the walls of the cylinder followed the exponential law of compression, and if the temperature of the gas were the same as that of the cylinder at the same heights in the stroke, the mean temperature would be easily found. We would have

$$\tau_2 = \tau_1 \left(\frac{v_1}{v_2}\right)^n$$

and since the valve opened at three-fourths the stroke, we have

$$\frac{577}{518} = \left(\frac{1}{\frac{1}{4}}\right)^n;$$

$$\therefore n = 1.114,$$

hence, mean temperature =

$$\left[\int_{\frac{1}{4}v_1}^{v_1} \tau dv + \tfrac{1}{4}\tau_2 v_1\right] \div v_1 = \frac{\tau_1}{0.923}\left[1 - \left(\frac{1}{4}\right)^{0.923}\right] + \tfrac{1}{4}\tau_2 = 88^\circ \text{ Fahr.}$$

nearly.

It is certain, however, that the average temperature will exceed this; it will be nearer the higher temperature than the lower.

Since it cannot be determined it will be necessary to assume a value, and we will take 105° Fahr. This will be 47° Fahr. above the temperature of admission, and only 11° Fahr. less than the highest temperature. The pressure of the gas entering the cylinder was 28.88 lbs., and the pressure in the coils is assumed to be the same.

From the equation of the gas (*Transactions*, Vol. X., p. 640),

$$\frac{pv}{\tau} = 91 - \frac{16920}{\tau v^{0.97}},$$

we have

$$\frac{28.88 \times 144}{466} v = 91 - \frac{16920}{566 v^{0.97}},$$

from which we find

$$v = 12.00 \text{ cubic feet},$$

which is the volume of one pound.

The test lasted 11 hours and 30 minutes, during which time the average piston displacement was 20,177.7 cubic feet per hour, and the volume of the sealing oil was 143.7 cubic feet, leaving 20,034 cubic feet filled with the gas, so that the ammonia evaporated was

$$20{,}034 \div 12.00 = 1669.5 \text{ lbs. per hour.}$$

REFRIGERATION.

The liquid ammonia entered the cellars at the temperature of 67.4° Fahr.; but, after passing the expansion valves, the pressure fell to 28.88 lbs., absolute, the corresponding temperature being, according to Regnault's experiments, −2° Fahr., as found from the formula (*Transactions*, Vol. X., p. 632)

$$Com.\ log.\ p = 6.2495 - \frac{2196}{\tau},$$

in which τ is the absolute temperature and equals $461 + T$, where T is the temperature in degrees Fahrenheit. Hence the fall of temperature was 69.4° Fahr., during which operation heat was imparted to the cellars by the ammonia, the amount being

$$69.4 \times 1.08 = 74.95 \text{ thermal units.}$$

The factor 1.08 is the specific heat of the liquid ammonia at the temperature −2° Fahr. (*Transactions*, Vol. X., p. 645).

The liquid passed along the coils, absorbing heat from the cellars until it was evaporated at the temperature $-2°$ Fahr., after which the temperature of the vapor was increased to 35.9° Fahr. as it left the cellar. The latent heat of ammonia at $-2°$ Fahr. is (*Transactions*, Vol. X., p. 640)

$$h_e = \frac{5065.7}{778}\left(91 - \frac{16920}{\tau v^{0.97}}\right)$$

$$= 592.52\left(1 - \frac{185.93}{\tau v^{0.97}}\right)$$

$$= 592.52\left(1 - \frac{185.93}{459 \times 9.44^{0.97}}\right)$$

$$= 565.6 \text{ thermal units.}$$

The specific heat of ammonia gas being 0.50836 (*Relation des Expériences*, II., p. 162), the heat absorbed in raising the temperature of the vapor 37.9° Fahr. will be

$$37.9 \times 0.50836 = 19.27 \text{ thermal units.}$$

The heat removed from the cold room per pound of ammonia pumped will be

$$565.6 + 19.27 - 74.95 = 509.92 \text{ thermal units;}$$

and for the 1,669.5 lbs. compressed per hour, it will be

$$509.92 \times 1669.5 = 851311 \text{ thermal units.}$$

The latent heat of fusion of ice is 144 thermal units (*Phil. Mag.*, 1871, XLI., p. 182); but as it is customary to use 142 in these investigations, we will use the same number. This gives for the "ice-melting capacity"

$$851311 \div 142 = 5995 \text{ lbs. per hour,}$$

or 71.94 tons of 2,000 lbs. each per 24 hours. The average indicated horse-power of the engine was 91.13 lbs., so that the "ice-melting capacity" was

$$5595 \div 91.13 = 65.79 \text{ lbs. per hour per I.H.P.}$$

If 30 lbs. of steam were used per indicated horse-power per

hour, the "ice-melting capacity" was 2.19 lbs. per pound of steam. And if 3 lbs. of coal developed an indicated horse-power, then the "ice-melting capacity" referred to coal was 21.93 lbs. per pound of coal.

BOILERS.

The plant contained two boilers, but during the test one was blown off and the fires under it thoroughly banked; and as the two were connected, the empty one served as a reservoir for steam. If any heat was absorbed from the banked fire—or if more heat was thus absorbed than was radiated from this boiler—it was not known, and no account was taken of it. The efficiency of the boiler was so high on the first trial that it was considered advisable to test it again, which was done just a week after the test of the plant above described. This test was for 12 hours. The engine was tested with the boiler the second time, but not the compressors nor the ammonia. This test showed that the engine and feed-pump consumed 3.60 lbs. of Lehigh coal per I.H.P. per hour, when the average horse-power was 102.92. If the same rate of coal consumption existed during the first trial when the average horse-power was 91.13, then the "ice-melting capacity" per pound of coal consumed was

$$65.79 \div 3.6 = 18.27 \text{ lbs.}$$

And as one pound of coal evaporated 9.6 lbs. of water at 66 lbs. pressure, the "ice-melting capacity" per 10 lbs. of steam evaporated was

$$10 \times 18.27 \div 9.341 = 19.01 \text{ lbs.}$$

If the feed-water be 180° Fahr. and evaporation reduced to what it would be at 45 lbs. pressure, the "ice-melting capacity" would be 19.88 per 10 lbs. steam, provided 3.6 lbs. of coal would produce an indicated horse-power at this pressure.

With our present knowledge of the properties of ammonia, the efficiency of a plant circulating ammonia in the cold room cannot be determined as accurately as one circulating brine. When brine is used, the quantity circulated can be directly measured and its specific heat directly determined; while in the use of ammonia both these elements are computed, and are subject both to the errors involved in the hypothesis in the theory and the errors of observation in determining the elements used. These errors may conspire to make the results either too large or too small,

without any indication as to the direction in which the error may exist; or they may tend to counteract each other. On the general principle that with every transfer of energy some energy is lost, it may be claimed that a plant circulating ammonia in the cold room will be more efficient than if it circulated brine, the latter receiving its heat from the same ammonia; but this also is theoretical, and in practice involves other considerations. We only assert that the amount of refrigeration of a plant circulating a brine can be more *definitely measured.*

It is not advisable, except in cases of necessity, to combine two tests in one determination, although it is preferable to arbitrary assumptions of coal or steam used per horse-power per hour; but any objection to so combining them in this case will be removed by discarding the boiler test and all considerations of fuel and steam, and terminating the investigation with the "ice-melting capacity" per indicated horse-power per hour, which is found to be 65.79 lbs.

The result fell short of the rated capacity, partly because the required speed was not secured.

The following is a summary of the observed and computed quantities:

FUEL, FURNACE, AND BOILERS.

The boilers were double return flue, arranged to run, automatically, between 60 lbs. and 70 lbs. pressure (gauge).

Test May 22d, 1887, 12 hours:	
Fuel, Lehigh nut (anthracite), heat of combustion, from chemical analysis, B.T.U.	12229.6
Coal for 12 hours, lbs.	4422.
Wood for starting 377 lbs. 0.4 coal equivalent, lbs.	150.8
Coal, total equivalent, lbs.	4572 8
Unburnt coal, lbs.	125.
Coal consumed, lbs.	4447.8
Combustible, total, lbs.	3577.8
Coal burned per hour, lbs.	370.65
Heat in 360.65 lbs. of coal, 360.65 × 12229.6, B.T.U.	4532901.
Coal per horse-power per hour	3.6013
Furnace, grate area, sq. ft.	39.
Ratio of heating surface to grate surface	35.63
Boilers, heating surface, sq. ft.	1389.
Water fed per hour, lbs.	3559.7
Evaporation per lb. coal fired, lbs.	9.341
" " " " from and at 212°	9.957
" " " consumed	9.601
" " " " from and at 212°	10.226
" " combustible, from and at 212°	12.703

Average gauge pressure, lbs	66.
Temperature of fire-room, deg. Fahr	81.
Average temperature of flue boiler, deg. Fahr	319.28
" " " feed-water " "	180.2
Total heat in 3559.7 lbs. steam above 180.2° Fahr., B.T.U.	3661463.
Efficiency of furnace and boiler, per cent.. $\frac{366146300}{4532901} =$	80.7

ENGINE (CORLISS).

First test May 8th, 1887, 11½ hours. Second test May 8th, 1887.

Piston, diameter of, inches	32.
" stroke " "	36.
" speed per minute, mean, ft	194.988
Revolutions per minute, average, first test	31.720
Average indicated horse-power, first test	91.13
" " " " second test	102.92
Water consumed per I.H.P. per hour, lbs., second test	34.104
Steam per I.H.P. per hour, lbs., second test	25.79
Steam condensed in the engine, per cent	24.35
Average ratio of expansion	5.807
Steam consumption by the feed-pump per hour, lbs. (estimated)	50.25
Steam consumption by the engine per hour, lbs	3509.45
Total heat in 3509.45 lbs. steam above 180.2° Fahr., B.T.U	3609479.
Heat changed to work per h., B.T.U., $102.92 \times 1980000 \div 778 = 261930$.	
Efficiency of fluid $\frac{261930}{3609479} =$	0.0721
Coal per I.H.P. per hour for engine and feed-pump, lbs	3.60
Coal per I.H.P. per hour for engine and feed-pump, if 1 lb. had evaporated 10 lbs. water, lbs	3.45
Combustible per I.H.P. per hour, lbs	2.91

COMPRESSOR.

Number of cylinders, single-acting	2.
Length of stroke, inches	36.
Diameter of pistons, each, inches	18.
Area head end of pistons, each, sq. in	254.47
Average number of revolutions per m	31.720
Piston displacement per stroke, cu. ft	5.301
" " " hour, both, cu. ft	20179.
Volume of sealing oil per hour, cu. ft	143.8
Volume filled with gas per hour, both, cu. ft	20036.
Indicated horse-power, mean	76.0892
Heat eq. of work by compressor per hour, B.T.U	193645.
Efficiency of compressor from coal $\frac{193645}{4532901} =$	0.0407
" " mechanism	0.755
Temperature of ammonia entering compressor, deg. Fahr.	57.7
" " " leaving " "	116.1
Initial pressure, lbs. per sq. in., absolute	28.88
Terminal " " " " "	132.01

Refrigeration.

Pressure in cooling coil, lbs., absolute....................	28.88
Hence, temp. of liquid, deg. Fahr........................	$-2.$
Temp. of compressed gas, deg. Fahr......................	116.1
" " gas entering the compressor, deg. Fahr.........	57.7
Rise of temperature due to compression, deg. Fahr.......	58.4
Latent heat of evaporation of 1 lb. at $-2°$ Fahr.	565.6
Superheating in cooling coils, 37° Fahr., B.T.U..........	19.27
Fall of temp. of liquid in cold room, deg. Fahr...........	69.4
Heat imparted to cold room by this fall, 1.08×69.4.......	74.95
Heat removed from cold room per lb., $565.6+19.27-74.95 =$	509.92
Ammonia evaporated per hour, lbs......................	1669.5
" " " " B.T.U....................	851311.
Equivalent in "ice-melting capacity" per hour, lbs.......	5995.
Equivalent in "ice-melting capacity" for 24 hours, tons (each 2,000 lbs.)......................	71.95
Equivalent in "ice-melting capacity" I.H.P. per hour, lbs.	65.79
Equivalent in "ice-melting capacity" per lb. of coal consumed at 3.6 lbs. per H.P., lbs......................	18.25
Equivalent in "ice-melting capacity" if 1 lb. coal had evaporated 10 lbs. steam at 66 lbs. pressure, lbs..........	19.01
Equivalent in "ice-melting capacity" per lb. combustible at 2.91 lbs. per I.H.P., lbs..........................	22.26
Equivalent in "ice-melting capacity" per lb. ammonia evaporated, lbs........	3.59

Efficiencies.

Efficiency of furnace and boiler (see above)..............	0.807
" " steam utilized by engine (see above)........	0.0721
" " engine referred to coal.....................	0.0582
" " compressor referred to engine (see above)....	0.755
" " " " " coal....$0.755 \times 0.0582 =$	0.0439
" " refrigeration referred to compressor.. $\frac{851311}{193645} =$	4.39
" " " " " I.H.P. of engine...	3.31
" " " " " boiler. $3.31 \times 0.0721 =$	0.238
" " " " " coal...$0.238 \times 0.807 =$	0.192

that is, for every thermal unit in the coal there was abstracted 0.19 of a thermal unit from the cold room.

CCCXC.

WORKING OF RAILROADS BY ELECTRICITY.

BY WILLIS E. HALL, ALTOONA, PA.

(Member of the Society.)

ALL innovations are taken up and accepted but slowly, and it is generally only through the continual driving of stubborn facts that we are willing to step (and then often reluctantly) from the beaten paths and ruts of our ancestors. All of this is better than to inaugurate a revolution, or attempt to divert a mass of moving bodies from one direction to its opposite, or in any other way which would prevent the operations of our every-day life from continuing with undiminished vigor and efficiency. But if once we stand still we become stagnated, and while nothing is to be gained by stopping or visibly checking the maintenance of an attained condition, yet there is no reason why development and maintenance should not go hand in hand, to the advantage of both sides. So when people predict changes and advancements of a few years hence, we must not reject them with a feeling and criticism that such would not be applicable to our "working of to-day," but rather stop to consider if our present methods could not be gradually altered to accept the new plan, provided it promised any better or more satisfactory results.

With this as a preliminary, it is asserted that the day is not far distant when we will see the railroads generally known as steam lines run by electricity from a central station. This is not, doubtless, the first time such an assertion has been made; but I have not as yet seen any discussion of the subject which has more than insinuated the advantages which would result in the operation of railroad lines by such a concentration of power. From a lack of time no attempt is made to analyze the two methods in a mathematical way, but rather it is aimed to point out some of the many advantages which the use of electricity allows, and which will inevitably result in operating a long line.

With this object let us take a few of the changes which the

substitution of centralized electricity would inaugurate when used in place of the present system of locomotives.

One very important gain would be the concentration of the power at one point, for a given length of line, into a few cylinders, instead of working it in a number of isolated engines where the insulation is poor and the chances for condensation the best. A number of central stations—located, say, at a distance of 30 to 40 miles apart—could be run by large, powerful engines, and the expansion of the steam worked at an economical point by better and more mechanical means of cut-off than can be obtained in the present type of locomotive construction. With locomotives the ratio of expansion at low speeds is correspondingly poor, nor can we expect to make an efficient engine of it, except in one working condition which is dependent upon the concurrence of so many variables that the engine is never worked in that ratio for any length of time. With stationary engines, however, the case is the reverse, as such are designed to work at a constant speed, and if properly proportioned would be utilized at the highest grade of expansion consistent with economy. This point would not vary much, as experience with hydraulic and electric plants would indicate. The fluctuations in the case of a railroad line would probably be even less, as the working is generally uniform throughout the twenty-four hours of the day. It would be a case of a properly loaded automatic cut-off engine against the equivalent of a similar engine vacillating between an over to an under load, and such range of the broadest nature. We must not lose sight of the fact that the question of keeping the line clear is properly held as of higher value in a closely worked system than is the consumption of coal which may result per the horse-power which the engine is to develop to reach its destination as scheduled. In furtherance of a reduction in coal consumption, the use of condensing—and possibly compounding—at a central station would raise this part of the working of the line to the highest grade of efficiency.

In experiments recently made it was found that an electric motor could climb a grade of over 15%,* which is far beyond the point of adhesion of locomotives; so that in this direction we could look for a marked improvement. An analysis of the

* In the paper as distributed in advance of the meeting, this figure was 50% by error. So much of interesting debate hinged on this point that it was not edited out with the correction, but this foot-note is inserted.

conditions in the two cases will disclose the reason for this, and no doubt all who have had any experience with engines which were overloaded or which slipped their drivers easily will appreciate its importance.

The question of attainable speed enters as a factor, for the speed to which an engine can safely be driven is known to have its limits—and which, to all appearances, we are now closely approaching. The piston-speed of an engine with 24 inches stroke and 68 inches diameter of driving-wheels, travelling at 60 miles per hour, would be about 1,400 feet per minute. An increase in the diameter of driving-wheels, with the object of decreasing the number of strokes, makes the engine correspondingly weaker, so that two sharp horns of a dilemma are placed before him who attempts to design an engine to haul the increasing weight of trains at a high speed. The questions of parts and velocity of steam are mentioned in passing. The resistance of trains is now quite positively known to increase about with the square of the velocity which would enter as a function in the power to be given to a motor to drive a train at a desired speed. With such a means, however, the speed is limited only by the power which is given in its design, and is determined more by what the conditions of the service will stand.

With electric motors it would not be necessary to have track-tanks and water stand-pipes distributed closely throughout the line, which means considerable to those who are acquainted with the attention and repairs (especially through the winter season) which such arrangements demand. The delay, too, where freight engines are not equipped with water scoops is apparent; and even this latter method of filling tanks is beginning to show its effects upon the schedule which it is possible to make on lines which are worked closely to their limit. Nor would it be necessary to carry the dead weight of tender with contained water, which is hardly as easy an accompaniment as its name would imply. In this connection it might be well to mention the annoyances from cleaning fires, as is required in freight service where the division is a long one.

The experience with the centralization of power where large hydraulic, electric, or pneumatic plants are in operation is that a greater amount can be supplied than it is necessary to develop at the station—that is, where there is much division (such as would be the case when the power is distributed through com-

mercial districts and divided into small parts) a 50 to 60 H.P. plant can take and supply satisfactorily about 100 H.P. In railroad work such a ratio could hardly be looked for, as the number of trains would not be as large as the division where the power is distributed for mercantile and commercial purposes; but a reduction of some 25% can safely be counted upon. The reason for this is evident, as it never occurs that all the power will be used simultaneously which each division is capable of exerting. With railroads, too, it will not occur that all trains would be exerting their maximum power– such as, for instance, climbing heavy grades—at the same time.

The best kind of mechanical ingenuity and high efficiency of any mechanical design which is made to accomplish a given object are dependent, principally, upon reducing the number of parts which it contains, so long as the desired result is obtained. Multiplication of parts increases the number of pieces to wear and consequent repairs, as well as the chances of failure from breakage. No argument is necessary to indicate the advantage which electric motors would have over the present design, or over any other design, of locomotive where all the requisites of an engine must be incorporated in so many isolated places. The failures from leaky flues, broken eccentric strap knocking a hole in the fire-box, blowing or knocking out cylinder-heads, and the multitude of accidents which are happening every day on railroad lines, would be decreased to a marked extent. The reduction in the internal friction of the driving mechanism is also apparent.

No comparative mention has been made of the cost of repairs, the too large percentage of power which is idle to have this work done, the reduction in the number of motors required to make the same train mileage (due to more uniform and consequent higher average speed, resulting in a reduction of time to go over a length of line), together with the loss of coal from irregular working of engine and boiler where the line is undulating, as is the case with all to a greater or less extent; also the attention and care which a large number of isolated boilers demand to keep them in a safe working condition, as well as the rapid wear of machinery where it is exposed to out-door influences, such as dirt and the elements, as is the case with locomotives. In fact, the elimination of such details could be extended almost indefinitely, and in them there would appear visions of a removal

of a mass of the little annoyances from the shoulders of those who are now held responsible for the maintenance of this accumulation of complications.

The loss in electric transmission has not been neglected, which in a station controlling a line of, say, 30 miles would, at the present state of the science, amount to some 50%—which includes loss in dynamo, line, and the loss from an average working of motor. This, together with the cost of necessary plant as capital, comprise the two main objections against the introduction of electricity for transportation purposes.

No attempt is made to take up the advantages or disadvantages (as shown by mathematical calculation) resulting from such a system, but it is merely desired to mention some of the many practical points which would be met, eliminated, or improved upon by the substitution of electricity to general railroad working. Nor is it that its introduction is anticipated within a year or two; but we cannot but acknowledge that the application of electricity is becoming more general, and, from the rapidity of its development, its use for such purposes is hardly more distant than the most sanguine of its advocates would predict.

The combination of electrical with mechanical engineering will bring about as much of a revolution in the future as it has done in the past, but in all its applications we must expect to see it creep before we may see it walk. A more thorough intermingling of the mechanical, however, would hardly be a detriment to much of the so-called electrical engineering.

DISCUSSION.

Mr. Oberlin Smith.—I rise not so much to discuss points which are prominently brought out in this paper as to speak of one point which is perhaps only hinted at. There is a certain engineering monster looming up in this country of late known as the electric locomotive. He is not mentioned by name here, and I do not really know what the author's views about him are. I imagine, however, from a passage in paragraph third, on page 3, where he speaks of one advantage in electric railroads being the saving of the dead weight of the "tender," that he intends to use a locomotive. This statement he amended by adding that another great saving would be in the dead weight of the locomotive. I know that it is quite common, in speaking of the future of electric railroads (which, by the way, I thoroughly

believe in and hope to live to see take the place of steam railroads entirely), to assume that there must be a locomotive at the head of the train, drawing the cars. I do not know from where the idea comes, except from the conventionalities of railroad men—a feeling, perhaps, that it would be too radical a change to take away from the train so useful a factor as the locomotive. But to me it seems a most absurd and ridiculous superfluity. Of course, with our present steam system, the running of this enormous mass of extra dead weight is the only thing to do. If we could have a more than ideally perfect rotary engine, with no more machinery about it than a grindstone or a coffee-mill (which describes very well the best electric motors of to-day, and will describe still better those improved ones which we hope to have in the future), and if, furthermore, we could pick up our steam along the track by simply running a little tube down to a cheap sliding contact or something of the kind, it would then be practicable to put these little rotary engines on every car-axle, or at least on one axle in each truck, and run our cars independently.

The steam locomotive has developed by evolution, and is, of course, a wonderful and beautiful machine, probably the best we could have attained to so far, and doubtless destined to still greater attainments. But it has great disadvantages in the dead weight due to carrying its boiler, fuel, and water, and in its reciprocating motions, none of which we can get over; and, too, we have certain limits of speed, owing to the reciprocating motions and the difficulty of getting steam fast enough through the ports, etc. With electrical propulsion we have no such limits, and I do not know where the speed limit may lie if we can contrive proper safeguards against derailment and collisions. Unquestionably, to my mind, the future electro-locomotion will show a motor on every axle, or at any rate upon two axles of each car, and every car running as a unit, in which case they can be run coupled together in a train or not, as may be convenient. In some cases, as on the New York elevated railroads, they can be run independently at times in the day when travel is light, and coupled when the line is busy. If thus run at close intervals much passengers' time would be saved. One of the great beauties of electric locomotion is that it is under such perfect control in regard to the amount of power used in proportion to the work to be done. According to many

indications we shall, after a while, be able to save the power used in braking and restore it to the line, although this scheme has not been perfected as yet.

In general, I do most emphatically wish to combat the idea that we must carry the enormous dead weight of a locomotive for absolutely no reason. We have the weight of the cars, plus the passengers or freight, for purposes of traction, and this is more than abundant, even if we make our cars in the future of lighter materials—as aluminium, steel, and what-not.

In speaking of the lightness of the future conveyances by rail, I believe we shall not only use steel and aluminium, but paper, India-rubber, and other fibrous substances, which will give us remarkably light cars, far beyond anything we can now speak of practically. Just as a wheelbarrow is to a bicycle, so will these present clumsy cars be to the future ones. But leaving this out, and speaking again of the electrical question and the fact that we shall have abundant traction in the cars themselves, I think we should set our minds against this idea of a great big motor car, loaded with several tons of ballast to give it traction, at the head of a long train of cars, and a man mounted on it to manage the electric switches. It is following in the paths of steam locomotion, which are all right in themselves, but are totally wrong in the new field of electro-locomotion, where many of the conditions are so entirely different.

Mr. Fred. A. Scheffler.—In reading over the article which has just been presented by Mr. Hall, I am tempted to give a few points in regard to the cost of installing a railroad plant such as the author of the paper thinks will be required in the near future, and at the same time I desire to state that it is to be regretted that Mr. Hall did not spend some little time, before writing his paper, in looking up the matter of electrical data as far as railroads are concerned, and the necessary cost of installation of the generating stations at a distance of not more than 30 miles apart.

The ordinary locomotive of to-day, when running at the average rate of speed and hauling the average load, does work of not less than 300 H.P. Supposing the station to be built to have a capacity, to each 30 miles of section, of not more than one locomotive, the capacity of the generating station would necessarily have to be about 360 H.P., as the best forms of motors at present have an efficiency of only

about 80% and 85%. It will require, therefore, for the generating station, a 360 H.P. engine and sufficient boilers for operating same. The cost of the station can be summed up about as follows:

One 360 H.P. engine	$10,000 00
Battery of 360 H.P. boilers	7,000 00
Station building	5,000 00
Steam connections—pumps, feed-water heaters, etc.	2,000 00
Generating dynamo	12,000 00
Electrical station appliances, etc.	1,000 00
Making a total of	$37,000 00

The foregoing station outfit will not permit of having an auxiliary plant of engine, dynamos, and boilers, which ought really to be included in the outfit, so that in case the engine or dynamos or boilers should at any time become inoperative, there would be another set of appliances to operate the railroad with. An auxiliary set of appliances would almost double the total cost of the station.

For operating expenses of the station we have the following:

5% depreciation on engine and boilers	$835 00
4% depreciation on dynamo	480 00
6% interest on the station plant	2,220 00
2 firemen, night and day	1,440 00
2 engineers, night and day	2,000 00
2 laborers, night and day	1,080 00
Maintenance, such as oil, waste, etc.	300 00
Making a total of	$8,355 00

per year, not including the cost of coal.

In order to ascertain what the cost of coal would be, we will suppose that the total power is required for only one-half the time, and one-quarter the power for the balance. This will give us 2,774 tons of coal per year required. This assumption is based on the supposition that the engine will require 3 lbs. of coal per horse-power per hour. If it was a condensing engine, of course the assumed rate of coal per horse-power would be too high; but I should not think it would be advisable to use a condensing engine where the power would be so largely variable, and water along railroad plants is not always available for condensing purposes. Basing the cost of coal at $3 per net ton, the expense per year for same will be $8,322; adding this to the

operating expenses of the plant at the station, we have a total of $16,677.

So far we have only taken into consideration the necessary cost of operating expenses of the station plant. We have now to consider the erection of the necessary electrical circuits along the line, the depreciation of same, and interest on the cost of same. Supposing that the pressure of the terminals was fixed at 2,000 volts, in order to deliver 300 H.P. to the motor it would be necessary to carry 112 ampères on the line, and allowing 20% drop in the pressure, by simple matter of calculation we find that the conductors will have to be of an area equal to a copper rod 1 inch in diameter. The pressure of 2,000 volts is altogether too high to think of using in a safe manner the rails for the return circuit, and a copper conductor would also have to be used for this circuit, making two lines of copper conductors 1 inch in diameter for the outgoing and return circuits. If the circuit is 30 miles long, it would require the enormous quantity of 476,784 lbs. for each conductor, and the cost of copper wire at the present time being 17 cents per lb., the amount required to pay for the copper wire would be $162,106. The interest on this outlay at 6% would be $9,726. As an overhead circuit would be the cheapest possible way of conveying the current to the motor, and as the wire is of such an enormous weight, it would not be well to have the poles more than 50 feet apart. This would require 6,300 poles. Basing the cost of the poles and erecting same at $2.50 each, we have a total of $15,750. Erecting the line would be charged at $150 per mile, and this is a very low figure; this would give us $4,500, making a total cost for the line of $20,250. The interest on this at 6% would be $1,215. Continuing the statement of operating expenses, it will be necessary to add the interest just given to same, and also the depreciation of the line: putting this latter at about 2% (which is very low), which would be $3,627, we have a sum total of $31,245 as the actual total annual cost of operating expenses of the station plant, and interest, depreciation, etc., on each section of 30 miles.

It will not be necessary to take into consideration the cost of operating the motor, as this would be about the same as the cost of operating the steam locomotive, and the first cost of same would be about the cost of a first-class locomotive. The only comparison which should be made between the operating

expenses of the steam locomotive and the electrical plant would be that of the difference in coal consumed between the station and the steam locomotive, and possibly the cost of repairs of the motor compared with that of the locomotive. I have not the exact figures of the coal consumed by the ordinary locomotive, but I do not think it is any less than 8 lbs. of coal per horse-power per hour. This being the case, and the locomotive develops 300 H.P. for 12 hours and one-quarter of this horse-power for the balance of the day, we would have per year a consumption of 6,570 tons; this, at $3 per ton, is equivalent to $18,710. Deducting the same from the cost of total operating expenses of the electrical station, we find that the difference is in favor of the steam locomotive to the extent of $12,475 per year. I do not think there would be any difference in cost of repairs between the electrical and steam locomotive. The total cost of the station, copper wire, and pole line complete would be $219,856. This amount of money would buy *about* 20 *new locomotives*.

The foregoing estimates are all based on inside figures, and if the railroad was 600 miles long, it can be readily seen what an enormous expenditure for electrical purposes would be necessary, and comparing this with the cost of the present steam railroad would be sufficient to upset any calculations in the line of electrical railroads.

I have selected a voltage of 2,000 as a basis for estimating the cost of the line, simply because it would require *extraordinary care* in insulation for a higher potential where the bare conductors, connections, insulating material, etc., would be subjected to all kinds of rough weather. If it were found that 4,000 volts could be satisfactorily insulated (and this has been done with *covered wires* in a few special cases), the first cost of line equipment would be very largely reduced, to, say, about one-half the above estimated cost, thereby reducing the yearly expense of operating by decreasing the interest and depreciation of the plant. As far as safety to human life is concerned, I do not believe that 4,000 volts would annihilate more completely than 2,000 would, but the chances for leakage of current at the higher voltage would be much greater. This would mean greater coal consumption and reduced efficiency of the electrical apparatus.

The operation of the plant at full power for one-half of the time involves, of course, the use of several motors, which has

not been considered in the matter of operating expenses. Such consideration would not make any appreciable *difference* in cost of operating expenses, because there would necessarily be the same amount of expenses in the line of engineers, firemen, and depreciation, etc., on the locomotives as there would be upon an equivalent number of motors.

A statement made by Mr. Hall on the top of page 3 is as follows:

"In experiments recently made, it was found that an electrical motor could climb a grade of over 50%, which is far beyond the point of adhesion of locomotives." I would like very much to make an inquiry as to the facts of this statement. It would be very interesting to the members to learn what motor it was, and how long the grade was, and what speed was attained when making this remarkable climb. It is well known, of course, that there is some very peculiar action between the tread of the wheels on electric street-cars and track whereby they can ascend a higher grade than any other kind of motive power, except, perhaps, the cable system, but I am very loath to believe that an electric motor could ascend a grade of over 50%.

It is safe to say that unless there are some radical changes made from the ordinary design of electric generators, and in the method of transmission of the current to the motor, as well as a motor of greater efficiency, we will never see electric railroads operated in this or any other country.

I venture to make the broad statement that *if steam transit would be permitted in our cities, the electric railroad business would not stand any kind of a chance for competition.*

Mr. Hollon C. Spaulding.—It must seem, in view of the statements which have just been made, that the owners and stockholders in the three hundred electric railways now in operation (in the United States alone) must be on the verge of ruin. But "ignorance is bliss," and at the present time they seem to be running along smoothly, and to have no difficulty in earning dividends. It may be out of the question to think that electric railways of a number of miles in extent can be operated successfully with the present method of transmission, although the present generators and present motors have an efficiency far beyond that of most steam apparatus to-day; and owing to the absence of reciprocating parts and all unnecessary weight, the efficiency of the electric locomotive or electric car is much

beyond that of the steam locomotive, whether you run it by throttle or by reverse lever. Now, it is useless, as I said, to think of running long roads by electricity by the present methods, just as it would have been useless in the first days of steam-engines, and when it was considered unsafe to use a pressure of more than 20 or 30 lbs., to get the results which we to-day get in stationary and locomotive engineering and by high pressures and condensation. But just as the next step after ordinary telegraphy was the telegraphing from moving trains without any contact whatever between the circuit which was actually used to send the message and the circuit acting at the point of delivery, so, it seems to me, there is no question that the next step, and the step which will solve the problem of electric transmission for railways, will be the use of high potential current along the track and the use of low potential motors with an induced current. This idea, as you know, has been utilized in lighting with signal success, and there is hardly a city in the country to-day but what is using the alternating system of lighting, in which there is no mechanical connection whatever between circuits on which the lights are placed and the high potential circuit which runs out of the station, and which is insulated so as to be practically without danger.

Regarding the grade which is mentioned in this paper, I am very glad that has been brought up, and I should like to have some explanation on that subject. I have not known of any tests which have given any such grade as that, although there is a slight percentage in favor of locomotives or cars in which the current is transmitted to the motors through the contact between the wheels and the rails. The amount of this additional traction has not been fully determined, but in such a case as that under discussion I may say that it seems to me too small to be taken into actual account. As far as the tractive force of electric locomotives is concerned, I do not think we should claim anything more than the results obtainable by steam, but I want to emphasize a little more the entire absence of reciprocating parts in electrical apparatus for propulsion. That has been touched upon as regarding the simplicity and consequent efficiency of the machines themselves, but mention has not been made, if I remember correctly, of the effect on the track, and on the bridges and structures over which the locomotives are to pass. In case of the application of motors to the separate

trucks, the absence of a concentrated weight is worthy of some consideration.

Regarding the speed of the electric railways in the United States to-day, the motors are usually designed for from 18 to 20 miles an hour; not on account of the limit of the electrical apparatus, but on account of the municipal ordinance limiting the speed of such vehicles in the streets. The higher the speed in an electric locomotive, within reasonable limits, the greater the efficiency, for the gearing on the cars is ordinarily about 16 to 1, so that the greater the speed the less the ratio of gearing.

One point has not been mentioned, and that is the opportunity which is afforded for utilization of the immense water powers which are going to waste day by day. In New England, in the South, and in the West there are enormous natural powers, as you all are aware, some of which can be and will be utilized by factories located on the spot. But some of these water privileges are so situated that factories cannot advantageously be located there. The interest in this matter is growing very rapidly at the present time, and some roads (notably one in the South, which will be about 200 miles in length) will undoubtedly within the next few years be operated by electricity, and that by the much-abused present method, which uses not over 500 volts—a perfectly harmless current.

Mr. S. J. MacFarren.—It is with some diffidence that I rise, in an atmosphere so permeated with steam-locomotive sentiment, to defend electrical methods. But before our worthy president made his point, which went to the root of the whole matter, I decided to get up and compliment the writer of this paper and congratulate the Society on this commencement of the study of a subject which is growing faster than perhaps many of us realize. I consider this both an interesting and suggestive paper, excellent in tone, while its moral—that more admixture of mechanical talent with so-called electrical engineering methods would be an excellent thing for all concerned—is truth itself.

I have some experience in business connected with electric propulsion, and I want to make the statement that the amount and quality of the quackery which has been perpetrated on the American public in the way of so-called "electrical engineering" is almost impossible to realize; and you would call it a fish story if I were to tell you some of the things to which I am a competent

witness. Mr. Oberlin Smith made an excellent point, namely, that no practical man would think of following the methods laid down in Mr. Scheffler's paper and confine himself to a locomotive having the ability to draw 30 or 40 cars, when he can not only get rid of the dead weight of the tender, but of that of the entire locomotive (by dividing the power among the cars).

I would be glad to show Mr. Scheffler, who is my townsman, an electric road where an independent motor of 12 tons' weight hauls a 16-foot ordinary car and load up a $15\frac{48}{100}\%$ grade, as certified to by Mr. Wilkins, of the P. R. R., by adhesion alone. It is practicable only with a clean rail and fair electrical conditions. I am inclined to think that the statement in the original paper refers to the experiments of Mr. Leo Daft, and perhaps that 50% may be a misprint for 15%. There is no question whatever, as the gentleman from Boston pointed out, that the passage of the current from the wheel to the rail, in what is known as the ground-circuit method of operating an electric road, does add to the adhesion for traction; whether this arises from any electric or magnetic cause (as claimed by Mr. Ries, of Baltimore), or whether it is from such causes as the mere heating effect, which would tend on a wet day to dry the rail quicker, I do not know, but the fact is there. The $15\frac{48}{100}\%$ grade I speak of is on the Pittsburgh, Knoxville and St. Clair Electric Road, and no one would claim that any known motor without the aid of electricity would climb a 15% grade and haul a heavy load as this has done again and again. I have observed in Fourth Street, Pittsburgh, where the cable track and electric track touch each other, that there is from one to three hours' difference in the time of drying off the rail after a snow-storm or rain; and probably this is due to the escape of the current through the rail.

As to the logic of Mr. Scheffler's paper, it is an entire surprise to me in this latitude and in this company. It presupposes an impossible case, and builds up an elaborate argument, which falls to the ground when you consider the statement of Mr. Smith, that the proper way to operate an electric railroad would be to subdivide its power and not to put it all in one place.

I am glad the gentleman from Boston told you about that Georgia railroad; and with reference to this question of dead weight, which is a very important and interesting one, I want to

call your attention to a curious invention of Mr. John C. Henry, of New York, who is a pioneer in the electric railway business. This is a motor without fields. It has three armatures, each of which in turn acts as field for the others, and the result is to rotate the central axle upon which all are mounted. It is only an experimental machine as yet, but Mr. Henry has been very successful in mechanical devices, and I consider him the best mechanician among the electricians of the country. My experience has been that the electricians, so called, are not mechanics. A system which is exporting motors to-day has been pronounced a mechanical failure in a careful report made by experts for a syndicate by which I had the honor to be consulted—has been so pronounced while acknowledged to be electrically the most perfect of all present systems. These experts state that the gentleman who gives name to this company, and whom I consider has worked out the mathematics of electricity better than any competitor—that he so exhausted his brain on the electrical features as to leave nothing for the mechanical details of the problem, and has not yet reached a place where he can guarantee (or name) any sum which will cover the cost of repairs to the gearing and mechanism of his motors. To illustrate how far expert prejudice can go, I want to say that I was asked not long ago how to avoid the "gallop" of cars on an undulating track. I said that if pivot trucks of four wheels each were too costly, the next best thing was a certain American gear, long on the market, and which admits of increasing the present wheel base (of 6 feet) to 9 feet. The capitalist repeated this to the superintendent of his home road. I was sent for to repeat the story to this gentleman and give the maker's name. He said: "There is something very strange about that. I have been buying car wheels from the firm for nineteen years and never heard of this." I took him three blocks from his office and showed him the gear in operation under a car, and he repeated his statement that in all his street-car experience he never heard there was any such gear. Three days later the superintendent, the capitalist, and three costly experts met in an interior city of New York to decide upon an equipment of an electric road. My statement was there repeated. One of these experts stated in the presence of the committee assembled that there was some mistake about this 9-foot wheel base, and that it was a practical impossibility for a 9-foot wheel base gear to go through ordinary street-car

curves. I told them to write to Mr. C. B. Holmes, of the cable road on the South Side, Chicago. They did so and learned that he commenced to use this gear when he had a horse road ten years ago; that he now had 775 sets in use and does not want any other gear. The Third Avenue Street Railroad in New York was using the same gear practically at the door of the expert who said it was impossible for that gear to work. I want to apply that to Mr. Scheffler's case. He has proved here by a very skilful argument that it is impossible to operate a railroad by electricity. But it becomes constantly less safe to define possibility. I am really sorry that gentlemen here show, if you will allow me to use the expression, the prejudice in favor of steam against electricity that appears here. Electricity is in no sense a rival, but in every sense an aid, to steam. I am not an electrician at all, but I tell you, you cannot afford to disregard electricity, because it will come without you, unless you think better to have it come with you, and any of you who look into it will agree with the conclusions I have drawn here, and will be astonished to find how much ignorance there is on this great topic. There never was an industry commanding the amount of investment the electric railway has to-day which was guided by such ignorant advice as it is dependent upon. No more inviting field than electric propulsion opens for mechanical engineering, since at least 90% of the features of so-called electric "systems" are *mechanical* features.

Sir Wm. Thompson has well said that the mechanical engineer needs to add but one-tenth to his knowledge to include the electrical field.

Mr. E. P. Roberts.—In connection with the remarks of previous speakers, and from the standpoint of an electrical engineer, I believe it very advisable that mechanical engineers should have a knowledge of electrical matters. An example of this is the fact that a large mining machinery firm have recently engaged an electrical engineer.

With reference to large train units, Dr. Louis Bell recently published a paper showing that if on a certain road between Chicago and Cincinnati electricity were applied as the motive power, and the traffic and train units were maintained as at present, a rather high potential would be necessary, and the coal consumption would be about as at present with steam locomotives. In most cases, small train units would be advisa-

ble. For elevated railroads, Mr. Daft thinks it would not be safe to operate trains closer than at present, and therefore, in order to handle the traffic, the trains would need to be of the present capacity, and for such work he advises an electric locomotive. To refer to the figures of a previous speaker, that for a road of the character he considers a conductor of one square inch section would be necessary, and the cost of such a conductor would be prohibitory, Mr. Sprague, in a recent paper, states that the telephone system between New York and Boston uses copper wires of an aggregate cross-section 50% greater than this, and that if such cost is not prohibitory for telephones, it is not for fast passenger traffic if the demand for such traffic is as great as between New York and Philadelphia.

Between these two cities he calculates that to handle the present through passenger traffic a separate fast line would take about 3,000 volts if operated from a station at one end, but if operated from four stations distributed along the line, and using the three-wire system of distribution, that a potential of 500 volts and a conductor of one-inch cross-section would be sufficient.

This latter potential is the same as is generally used on street-railway lines; 2,000 volts have been used successfully on series lines.

Mr. Sprague makes some assumptions—such as electrical braking—which have not been, as yet, realized in practice, but which doubtless will be thoroughly worked out before long.

With reference to electric railways paying, it certainly is a fact that such is the case. I am informed of one road where they are carrying 3½ people per car mile, receiving five cents for each person. They are operating over twenty miles of double track, and are making money. Also of another road where the expense per car mile is less than eight cents.

Mr. C. J. H. Woodbury.—I would like to ask the speaker a question on that point, and that is if the expense of eight cents per mile refers to the whole expense of the corporation, or only operating expenses?

Mr. Roberts.—I was told by one who was supposed to know, and supposed to be trustworthy, that it included all.

Mr. Woodbury.—Interest on any bonds, or anything of that kind?

Mr. Roberts.—Interest on any bonds—everything of every nature that would go into expense.

(Later Note. I am since informed that the expense of eight cents per mile includes everything excepting interest on bonds, and that ten cents covers items of that nature.)

Mr. Jesse M. Smith.—Mr. Scheffler has made out a remarkable case for the steam locomotive as against the electric with stations along the road. He supposes, as I understand his remarks, that these stations are to be used at all times, whether there is a train on that particular section or not, and that each station is developing its full power at all times. Now, that is not the case. There is a continuous line running right through from one end of the road to the other, fed only at intervals, and if the supply of electricity which is drawn by the motors passing a particular section is not needed, that supply goes on to another section and helps that out; and it is a matter of practice in street railways that a great many more cars can be put on a line than is actually represented by the power in the station. That is, if the station is for a 1,000 H.P. there can be as many as 1,500 H.P. of motors put on that line without any detriment whatever, supposing that some of the motors are idle, while others are developing their full power. That one idea alone would cut down Mr. Scheffler's expenses one-half. I think it is a point which has been overlooked before in this discussion.

Mr. Chas. W. Barnaby.—In regard to the method of driving through the application of motors to the car axles, I have noticed, in late discussions of the subject in the electrical papers and conventions, a disposition on the part of some to advocate returning to the locomotive principle. I do not recall all of the arguments presented by the advocates of this system, but one reason given was that the fitting with motors of the extra cars used only for heavy traffic necessitated the investment of considerable capital, which must lie idle much of the time. By using a sufficient number of electric locomotives to take care of the ordinary travel with but one or two cars attached, the heavy traffic being provided for by attaching more cars to each of the regular locomotives, it was claimed that the cost of equipment was less than by the separate car method of propulsion.

As Mr. Smith has stated, driving by direct application of motors to the several car axles appears to be by all means the

correct principle, and it is to be hoped that no practical considerations will necessitate returning to the independent locomotive, as that looks like a step backward.

Mr. D. L. Barnes.—Perhaps the railway mechanical engineers are a little ahead of the electrical engineers in this matter. To my knowledge, in the last year three roads in Chicago have made estimates on the transmission of electric power for switching and carrying passengers in Pullman cars through the streets of cities, and upon application to electricians for stationary motors for the purpose they could get no reply which would give them any clue to the economy or desirability or operation of the necessary motors, none having been built large enough to perform the work.

Mr. Jno. M. Sweeney.—I probably have been placed in a position for the last two years to know something of the troubles connected with the electric propulsion of cars, because I have been interested in a street railway line and have been actively connected with its management. I think that in the consideration of the question before the body the fact is lost sight of that electricity is merely a ready means of transmitting energy instead of a belt or a rope or other methods which have been employed at various times. One of Mr. Scheffler's points—the last one which he made—struck me as rather erroneous; because he said that, under certain conditions, if steam cars were allowed to be run through the streets of cities, they would instantly supplant the application of electricity for that purpose. That can hardly be possible, because for street traffic a higher class of labor is required to take care of and manage the steam locomotive than to take care of and manage an electric motor, and it is that feature which makes electricity so particularly applicable to street traffic. You can pick up a class of labor and educate it in a short time to handle the electric appliance. Further (and it is the gist of this whole matter, to my mind, as it applies to long-distance electric transmission, or, rather, to replacing the locomotives in what are usually known as railways), that where you can concentrate the output of your plant and allow it to serve a large number of independent operating machines you have what is a desirable commercial investment, while if you had only one car to run or a lesser number of street-cars it would not become a desirable thing, for you had better put the balance or the whole of your expense on the one

or two cars or whatever it may be. It must be under a condition where it is commercially judicious to use the central station for the distribution of power over a number of cars.

There has been some comparison made between the reciprocating parts of steam locomotives and what is gained in electrical apparatus by dispensing with reciprocating parts. That is very true, but in the present state of the art, and the application of electricity to the propulsion of cars, it would be done at a very heavy expense in the gearing employed to connect the motor to the axle of the car, and that to-day is the greatest obstacle to be met with by one managing or attempting to conduct a street-car line driven by electricity. My first experience with it extended to what was then known as the Van De Poele method. It was at an early time in the introduction of electrical street-car lines—I think about the sixth road which was operated. The Van De Poele method consisted in placing a motor in the front end of the car, and connecting it to the axle with chains. Two chains were used, connecting from the counter-shaft of the motor down to the wheel on the axle of the car. It became evident in a little while that one of those chains was of very little use; that it was impossible to concentrate the strain on both of the chains or equalize it, and that one of them was merely idle. With that idea, I caused one of the chains to be removed, and ran the car much more successfully with one chain than it had been operated with two chains. The chain which we used originally was a chain with a roller between the links, working on a pin, and the stretch of it was very considerable, so that it was constantly increasing its pitch, while the wear of the sprocket-wheel diminished it, and we had the two things bearing one against the other. The cost of the chain was $1.60 net per foot, and about seven weeks was the life of it. We now use malleable iron chain, costing 60 cents per foot, which we run at least six months. There is too much disposition on the part of mechanical engineers to conclude that a requisite for electrical application is a lack of mechanical engineering knowledge. For a time I was of that opinion, and thought that it would be but little trouble to correct many of the weak places in mechanical application, but I have generally found that they are too closely allied with the electrical question, and that what would certainly be a successful device if electricity was left out of the question became anything else when connected with the electrical end. There is

a great field for mechanical engineers in working out the details of the adaptation of electricity. But in order to be successful the engineer must also know a great deal of the properties of electricity.

I think Mr. Scheffler's figures on the amount of wire, etc., required to operate a line, which he assumes, undoubtedly correct, and that to any such extent it is decidedly impracticable to operate the line ; but new methods of transmission, new conditions in pressures, etc., can be expected at any time, and may change a great many of the conditions. However, I am convinced that for long lines electricity can only be applicable to advantage where the unit of load to be moved is very small, say one or two cars in train. The advantage found entirely disappears when the line is extended, and the train unit increased, as if there were twenty or thirty cars.

Mr. Wm. Forsyth.—I should like to say a word on this question from a railroad standpoint. While we want to be as progressive as possible, and to go into the use of electricity, we will not be discouraged or terrified by the figures which have been presented by Mr. Scheffler. He has very properly called these inside figures, and the railroads will not use this kind of figures for their estimates. When they go into the wholesale use of electricity, they will manufacture their machinery the same as they now manufacture their locomotives—in their own shops—and the cost of that machinery will probably be about one-fourth or one-fifth of that now charged for electric machinery by electric companies. I have been using some electric motors, and in trying to use more I am met with this question, which is purely a commercial one : that of the cost of the machinery ; and in getting estimates for a 7½ power motor I found that figures from the various companies varied from $1,100 down to $400. Probably the $400 motor was as good as that they charged $1,100 for. So that, as I say, these figures which Mr. Scheffler has presented can be considerably reduced in the mere cost of the motor itself to start with. As our President has said, the motor is a piece of machinery which is about as simple as a grindstone or a coffee-mill, and instead of costing $12,000, as given in this estimate, I should expect to build a motor or dynamo of the same capacity for about $2,000.

The President has spoken about that monster, the locomotive, and the probability that such machines would not be used at all

when electricity should come into use. But there is one phase of railroading in which that would be used, and that is in switching; and I believe that the first use of electricity on a large scale on railroads will be large motors for switching-engines. In our switching-yards we have to employ a heavy locomotive, with its tank, an engineer and fireman, and a man at the back end for handling the push-pole, making three men, with a very large and heavy piece of machinery. Now, if we can reduce that to a small, compact, and heavy motor, with one man directing all the motions and the motion of the pole besides, it will certainly make an improvement, and it is one which I believe will be the first application on a wholesale scale of electricity to railroads.

The other question which the President mentioned was the use of a bicycle construction in cars. While it is very desirable to reduce the weight of cars, there is one objection which it meets with, and that is the matter of safety. When we obtain a speed of 50 or 60 miles an hour, any one who is travelling along at that speed would a great deal rather be enclosed in a good, substantial box than in a very light and flimsy one, so that in case the thing goes off the track he will be better protected. I believe, though, in the use of iron and steel where we have the maximum strength for a given weight of any material which it is possible to use for railroad cars, and that by the use of them we can reduce the weight considerably.

Mr. B. J. Dashiell, Jr.—I would like to go on record that there is no virtue whatever in passing an electric current through a wheel to a rail and thus increasing the adhesion. From experiments made by Mr. O. T. Crosby and myself at Jamaica, L. I., we found there was nothing in it. Our experiments were published about four weeks ago in the *Electrical Engineer;* Mr. Daft came out in the next issue and agreed in the main with our experiments which were published. Yet Mr. Ries, of Baltimore, tried to obtain results by passing a current through a wheel and rail, but found that he could accomplish nothing in this respect. I believe he then built a model, wound his wires upon the axle, magnetized the wheel, and got some fair results. The results obtained by him on the model were by far more favorable than those which could be obtained in actual practice.

Mr. W. O. Webber.—I think this discussion develops one thing, if no other, and that is, they are *all* right. I think Mr. Scheffler is not all wrong; and I do not think the others all right, and

vice versâ. This is proven by the fact that there are electric street railroads being operated to-day with commercial success. They have been operated this way for four years at least. Now, if this principle was absolutely applicable to the traffic on steam railroads, I do not think the American public are quite so slow-going that they would not have got at it in four years. I do not think Mr. Scheffler is entirely wrong, because there are certain items which have got to be taken into consideration in looking at it from the standpoint he did. One is this: you cannot operate a big trunk line by electricity in the same method by which you would operate a street railroad, with small cars running at very short intervals of time, without a great deal of preparation. For instance, you would have to have four tracks, without any question, two each way for passenger traffic at high speed, and two each way for freight traffic at slow speed; because you could not have on the same track trains running at one speed and other trains running at one-third of that speed. You have got to operate your freight trains as an aggregation of a large number of cars going at a pretty slow speed. As Mr. Forsyth says, you cannot do switching in this way without pretty heavy motors. I think the argument was very good on both sides. The proof is the absolute fact that you have got lots of electric street railways to-day churning out coupons, and you haven't got any big electric trunk lines.

Mr. Oberlin Smith.—I will not take up all the time I would like to take in answering the various points which have come up. But I want to say in answer to Mr. Forsyth, that when we run the fast trains and the light cars we expect to, we won't depend only upon a wheel-flange $1\frac{1}{8}$ inches deep to keep us on the track and out of eternity.

With regard to Mr. Scheffler's statement, I will just say that in his calculation of the amount of power required to run an electric train, he assumed the same power as for a steam train, and forgot to deduct the weight of the locomotive, which with many of our passenger trains probably averages nearly as much as the cars themselves. Take that off, and the horse-power and, consequently, the coal consumption would be reduced in proportion, taking a large percentage out of his cost figures.

Mr. Thos. R. Morgan, Sr.—The great difficulties with which some minds have to contend in discussing new powers, such as the electric, as in this case, comes from the education already

received, and an unwillingness or determination not to accept the new conditions until they have actually forced themselves into practice. With such minds every argument possible is brought to bear against its success, which, after all, in a general way is educational and desirable. Other minds are more hopeful, willing, and desirous to live and accept progression as it comes along. My disposition and experience make me hopeful and desirous to accept all progressive powers as they come along. In this spirit we have used electric power in our workshops in many and various ways, with universal success so far. We are becoming more and more friendly to it as we go along. I believe in its great future possibilities, even to railway locomotion for reasonable distances through auxiliary power stations, if such can be accomplished. The engineer and fireman, also the dead carriage weight of fuel and water for steam purposes, will be dispensed with. The minor difficulties now existing will, I believe, in the future be overcome to a considerable extent. We are running, and have been for about one and a half years, one each, ten, fifteen, and twenty-five tons overhead travelling cranes operated by electricity. They have given us no trouble whatever since we put them in operation. We have electric motors running complete lines of machinery and shafting with similar success, reducing largely the use of belting, shafting, and gearing. I would respectfully invite any member who is interested and may desire to see our practice to visit us at Alliance. We shall be pleased to show all who go the many evidences of what I say here. My desire in speaking as I do (based upon the experience we have already had with electric motive power) is to help prepare the minds of our members to look hopefully, as I do, to the possibilities of this great power in the future. Its superiority over any other power, in refinement, simplicity, and safety for operating many classes of machinery, we have established to our own satisfaction beyond a doubt. We have, consequently, reasons to believe and feel hopeful that even electric railways for long distances may be a possible success in the future.

While believing in the revolutionary displacing of some of the other powers by electricity, I do not want to be considered an enthusiast to such an extent as to say that electric motive power will be best for all places and under all conditions where steam, hydraulics, pneumatics, and gas are now being used, and are best suited for many purposes. These powers will also have

their special demands, as they have had, as varying locations, purposes, and conditions will favor either one or more of the powers in combination as the best and most economical; all of which in our business we use and furnish to others, in part or combined, to suit the wants or demands of our customers. We find the conditions are such in two different locations, even when the demands are similar, that it is wise to change the motive power to suit such conditions and locations. While I believe in the considerable advancement in electric power for locomotion and in our business for many purposes, I am confident all the other powers will also hold their own.

My desire in expressing myself as I do is to show my views and experience as favorable to electric motive power, and that I believe in it, but, at the same time, I do not lose sight of the other powers for similar and other purposes.

Prof. Arthur T. Woods.—The author states that "the resistance of trains is now quite positively known to increase about with the square of the velocity."

As reports of tests in this country and abroad—as, for example, tests on the Chicago, Burlington and Quincy Railroad, and the recently published indicator cards from a Worsdell compound locomotive, taken at speeds as high as 86 miles per hour—show that the train resistance not only does not increase as the square of the velocity, but is in some cases less at high than at low speeds, Mr. Hall will add greatly to the value of his paper by giving us data which will support his statement.

Mr. Fredk. A. Scheffler.—I desire to correct what seems to be the prevailing opinion of some of our friends who have partaken in the foregoing discussion of my comments of Mr. Hall's paper, in that my statements were with a view of creating a feeling that I am not in favor of electrical railways of any kind. I most emphatically desire to say that an opinion of this kind is entirely inconsistent with the statements I have made. I am particularly desirous of furthering any matter which pertains to the advancement and introduction of electrical work of all kinds, as there is no branch of the world's industry which is or will not be influenced by the electrical industry. Careful perusal of my discussion will show that in no case have I even hinted that I am desirous of crying down electric railroads, and I am extremely disappointed to learn that certain members did not give as care- attention to my remarks as I should certainly have desired.

My object in presenting the comments was explicitly stated in their first paragraph, and in endeavoring to show the cost of installing and operating one section only of such a railroad as Mr. Hall has suggested, I fail to find anything which would give rise to the opinion that I am personally against progress in this line. I, too, desire to see electricity supplant the present steam railroad, but *at present* cannot see exactly how this can be accomplished from a commercial standpoint. In this respect I concluded my remarks with the suggestion that it would be necessary to make a considerable advancement in the line of a better and cheaper generator, motor, and method of transmitting the current from the generator to the motor. Does this statement even suggest that I am *not* in favor of electric railway work ?

If the same progress is made in the next ten years as has been achieved in the past decade, there will be more of a possibility of our capitalists adopting electricity for steam in railroads.

Three different members have asserted that I have estimated too high by fifty per cent. in three different items. I do not desire that my figures shall be taken as absolutely correct, but I would like to have a comparison of cost by parties who are familiar with electrical and mechanical estimates, from a conscientious point, made upon the same basis which I have estimated. I shall be perfectly happy to acknowledge any errors which I may have made, and will be only too glad to cut down the apparently enormous first cost of even one section of the suggested railway plant.

A specially devised method of feeders would possibly reduce the cost of the copper circuit, but at present I do not see wherein even this could be accomplished, unless, perhaps, a three-wire system should be used, but which would involve a further complication of carriers, etc. One thing certain is that to carry 300 H.P. at 2,000 volts, with 20% drop, requires the same amount of copper for conductors, whether the latter be composed of one or is subdivided. I have taken as a basis the lowest possible generating capacity, and an increase of this simply means additional first cost, and some of our heavy locomotives surely develop a much higher amount of power at certain times than 300 H.P. I have also estimated upon only one motor being in operation on the 30-mile section, whereas it would not be a very paying road which would operate under such a condition. More than one

motor upon the section at the same time involves a still further increase of generating plant and conductors.

I have only presented the latter thoughts to show that I do not think my estimate too high.

*Mr. Willis E. Hall.**—The figures presented by Mr. Scheffler would appear, on their surface, to eliminate all chance of a successful railroad with electricity as the driving power. They but present the conditions which one would be apt to meet were he to start to-morrow and attempt to establish a line of some 30 miles in length, transmitting and using only appliances which are in successful general use at the present time. We do not know if Mr. Scheffler is a representative or an advocate of the high potential system of transmitting, but he will probably recollect and can review the somewhat unpleasant struggle which it was necessary to pass through before the proper authorities and the public were willing to accept high tension currents as a safe method of transmitting and distributing the power in question, aside from its economical value. Whichever side Mr. Scheffler may take in regard to low or high tension currents, he must at least acknowledge vast strides in this direction within the last few years. The jump from 2,000 or 3,000 to 10,000, and now to 20,000, as in a plant under construction in England and soon to be placed in operation, goes forcibly to indicate the sense of appreciation of the economical side of electric transmission.

In regard to the estimate which Mr. Scheffler makes for the driving power, if such an application of transmitting were put in play it is hardly likely any one would pay $10,000 for a 360 H.P. engine, nor would it hardly be expected a dynamo of the same power would be placed at a $12,000 figure. Mr. Scheffler is, I presume, well aware that with any commodity an increase in demand invariably reduces the cost to a corresponding extent, and also well aware that the electric motor business is now in its infancy, and the figures now charged are but temporary. The case cited by Mr. Forsyth illustrates the point in a very practical way. There has, probably, never been a new field into which the public can enter so freely as that of the manufacture of electric dynamos, motors, and appliances. It seems to be merely a question of evading patented details, which mechanical ingenuity so much succeeds in doing with devices

*Author's closure, under the Rules.

similar to those which are necessary to properly apply electric power. It is to be regretted that Mr. Scheffler did not go further and undertake to detail the cost of the operation of a steam railroad when locomotives are used in contradistinction to motors driven by electricity. It was at this point that the writer stalled, and must now confess the advantages to be gained are perfectly clear, yet of such a nature as to be evident only to those who have a daily acquaintance with practical railroad working.

It should be understood, of course, that it is not intended that the introduction of electricity for the purpose in question would be to the complete extermination of the locomotive, any more than we would expect or do find the use of the same power for lighting is to the utter exclusion of gas. It is apparent why the latter consumption has increased with the use of electricity for lighting purposes, and we would as soon learn that the easiest handled power is very apt to be made the most economical, in the same way that mankind follows the law of nature in endeavoring to find and introduce the line of least resistance in all work. A larger field would soon be found for steam than at present, while we would also be making a step toward further conviction that the least manual power called for is to the advantage and comfort of all concerned.

The words of Mr. Thomas Morgan, Sr., are particularly interesting and of the same tenor as when the writer suggested the use of electric motors for running overhead travelling cranes, and which he has since proven conclusively to himself and to others is the only proper method of driving them. The rather endorsing nature of Mr. Morgan's remarks, coming from a man of his experience, is encouraging to those of us who hope and expect to live to see the steam railroads of the country driven by electricity, or even some more mechanical means than at present in use.

The discontinuance of shafting and belting by motor transmission, which Mr. Morgan mentions, brings out another point which we are soon to see started—the use of individual motors to drive each individual machine.

The 50% grade referred to in debate was a misprint, and has been corrected to its proper figure—a 15% grade.

CCCXCI.

THE EFFICIENCY OF LOCOMOTIVES.

BY W. F. DIXON, PATERSON, N. J.
(Junior Member of the Society.)

EVER since Mr. D. K. Clark, by his masterly series of experiments forty years ago, demonstrated the existence of cylinder condensation, the object of by far the greater number of improvements which have been made in steam-engines has been to improve the distribution of steam in the cylinders, and to mitigate the losses arising from liquefaction during the earlier stages of expansion and re-evaporation during the later. Objections against the shifting link valve-gear, invented by Howe, but commonly known as Stephenson's, have been raised so often and urged so persistently, that many appear to take for granted that it is a wasteful and inefficient device. While it undoubtedly has faults, whose existence it would be absurd to deny, its many advantages would seem to be a sufficient guarantee of future use as extended as its past. The two principal charges against it are (*a*) the slow opening of the ports for steam admission, causing wire-drawing and loss of pressure, and (*b*) the early closing of the exhaust, creating an undue amount of compression.

A perfectly horizontal admission line and a sharply defined point of cut-off on an indicator card are, of course, desirable, but a considerable deviation from them is possible before a really appreciable reduction of the mean effective pressure takes place. The excellent means of balancing slide-valves now in use, unpatented and free to all, permit ports of large area to be employed, while the supplementary port of the Allen or Trick valve is a valuable aid in overcoming the fault of wire-drawing. It is, therefore, no exaggeration to say in 1890, as Mr. Clark did in 1852, that with a well-designed link motion of liberal proportions the objection of wire-drawing is of no practical weight. With regard to compression, there is no doubt that at times it does become excessive, considerably more than is necessary for the smooth running of the engine. Even

if we did not have the evidence of the indicator as to this, substantial proof is afforded by the fact that the main crank-pins of high-speed locomotives are usually worn more on the sides than on the top and bottom, the pin being on the centre when calipered.

The clearance spaces in any cylinder bear such a close relationship to the compression curve that they must enter into all discussions of it. In engines like the locomotive, where a single valve controls all the events of the stroke, it is undesirable to reduce the clearance spaces to less than 7 or 8% of the piston displacement, for at short points of cut-off the compression line would run up so abruptly as to reach initial pressure before the piston had reached the end of the stroke. From one point of view, the steam of compression should reach a pressure just sufficient to arrest the motion of the piston, crosshead, etc. Any increase above this detracts unnecessarily from the area of the indicator card, but does not diminish the efficiency of the engine. On the other hand, if compression is carried to initial pressure, none of the incoming steam is required to fill the clearance spaces and to raise the cylinder walls to a temperature as high as its own. This is indubitably the reason why the locomotive engine at high speeds and short cut-offs is as economical a machine as it is. The reduction of indicator card area caused by early exhaust closure, and the large clearance spaces found in locomotive cylinders, is often overestimated. A certain engine, with a 24-inch stroke and worked with 160 lbs. boiler pressure, has clearance spaces equal to 10% of the piston displacement. When cutting off at 6 inches it has been found necessary, to insure smooth running, to compress to 54 lbs. To do this, the exhaust must be closed at 15.28 inches on the return stroke. Suppose, now, it was possible to reduce the clearances to 2%, steam being cut off at 6 inches as before. To compress to 54 lbs. the exhaust must close only after 22.25 inches of the return stroke have been travelled. Fig. 188 shows the theoretical indicator cards which the engine would give in each case, and it will be seen that the M. E. P., when the clearances were 10%, is 94.6 lbs., sinking to 91.8 lbs. when the clearance was reduced to 2%.

If compression takes place suddenly, as is the case in cylinders having very small clearances, the heat does not have time to become equalized between the steam and the cylinder walls, and the power required for compression will be in excess of that necessary were the action to take place more gradually.

From the foregoing the writer has been led to the conclusion

that although the clearance spaces in locomotive cylinders are usually larger than desirable, there is a limit, pretty sharply defined, below which it is prejudicial to go in reducing them, provided the link motion is retained.

In cases where excessive compression has been found to limit the power of high-speed locomotives, a little inside clearance to the valves has been of great benefit in reducing it. This remedy might be advantageously used far oftener than it is, and would materially lessen the force of the objections erroneously urged against the link motion.

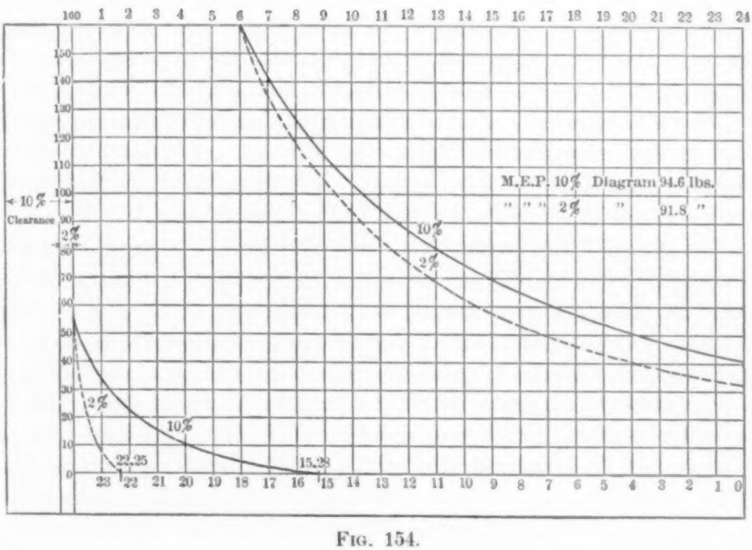


FIG. 154.

The influence of clearance spaces as affecting the nominal point of cut-off and the terminal pressure has been treated of so often that no mention need be made of it here.

To overcome the evils of cylinder condensation and re-evaporation, (1) steam-jacketing, (2) superheating, and (3) compounding have been suggested and tried.

That steam-jacketing the cylinders of a locomotive, taken by itself and not in connection with compounding, would result in some little saving may be taken as certain, provided sufficient means of draining were used. The application of jackets to the cylinders of an ordinary locomotive by Mr. Borodin, of Kief, Russia, showed a mean economy of steam of about 13%. This,

however, is probably much greater than could be expected in regular day-in, day-out service, and it is still further probable that the expense of constructing and maintaining the jackets would more than neutralize the benefit derived.

Superheating the boiler steam, to the degree of increasing its pressure, before admitting it to the cylinders has been tried several times in locomotive practice, and as often abandoned. The chief difficulties met with were: (a) The expense of providing and keeping up the superheating apparatus; (b) the difficulty of lubrication, owing to the vaporization of the lubricants at the high temperature; (c) the impossibility of preventing leakage at the stuffing boxes, all vegetable packings charring with the heat and becoming worthless. Nor is it likely that the metallic packings, now so largely used, would be able to overcome this last difficulty, for, with a possible exception or two, their prominent features are steel springs and white metal rings. The springs would gradually lose their temper and become useless, while the soft rings, fusible at temperatures of from 250° to 400° Fahr., would melt.

The Chicago, Burlington & Quincy Railroad Company, between the years 1870 and 1874, went fully into the subject of superheating. In these experiments the front tube sheet of the engine was set back into the barrel of the boiler some distance from its normal position. In the space thus left was fitted a separate cylinder, filled with tubes through which the products of combustion had to pass on their way to the atmosphere. The dry pipe entered this cylinder at the top, and the steam, after circulating among the tubes, and becoming superheated in doing so, passed to the cylinders of the engine through pipes connected to the bottom of the superheater. The resulting economy was, however, more than offset by the expense of keeping up.

The advantages claimed for compounding are two in number. First, economy in the use of steam and, consequently, fuel; and, secondly, a more uniform pressure on the cranks. The latter, although largely incidental, is held by some engineers, rightly or wrongly, to be the greater of the two, as a more uniform crank-pressure, followed to a logical conclusion, means greater durability of parts, and a possible lengthening of the life of tires, caused by the reduction of wheel-slipping.

It is tolerably certain, though, that were this to begin and end the advantages derivable from compounding, the continuance in favor of the ordinary single expansion engine, to the exclusion of

the compound, would be assured. The economy shown by the compound locomotive is usually attributed to the lessening of cylinder condensation, due to reducing the range of temperature per cylinder. When engines with well-protected cylinders are running at high speeds, it is hard to understand how condensation can take place to any hurtful extent. The writer has examined a large number of cards taken from locomotives, at speeds ranging from 40 to 65 miles per hour, but has been unable to detect any trace of either condensation or re-evaporation. Supposing, though, that some small amount of steam, say 7%, was condensed at each stroke, why expanding in two cylinders, one presenting twice the area of the other, should diminish it is almost beyond comprehension. At slow speeds, however, the benefit of compounding is beyond dispute, as witness the success of the system both in marine and stationary work; but the bulk of the compound locomotives so far built has been for express passenger traffic. If, then, the increase of boiler pressure which has almost invariably been introduced with compounding cannot be held accountable for the economy claimed, it would seem that Sir Frederick Bramwell struck the right key when, in a lecture delivered by him in 1877, he said: "There is no doubt that the double-cylinder engine is the engine of the present day. I do not think its economy really lies in its principle, but that its economy in practice arises from another thing altogether, and that is this: That by making a double-cylinder engine you put it out of the power of an ignorant engine-driver to do away with that which you want—high expansion. He must get high expansion; he is compelled to use it, whether he likes it or not. Whereas, with the single-cylinder expansive engine, he has the power to follow the dictates of his own ignorance; and, as a matter of observation, I have hardly ever seen such an engine left to the control of an engine-driver but it invariably worked at the lowest possible grade; and, as I have said, I believe that this withdrawal of control is to a large extent the secret of the success of the compound-cylinder engine."

Locomotive-engine runners are notoriously disinclined to run their engines with a full-open throttle, and to regulate the power of the engine by the valve-gear. But it will not do to make so sweeping an assertion about American runners as Sir Frederick Bramwell did about the English, for the design of the steam-distributing gear on many locomotives is such that the engines cannot be handled entirely with the reversing lever. When in compound-

ing we get an engine which *must* be worked expansively, irrespective of the shortcomings of the valve-gear, it is obvious that good results must follow.

The idea of applying the principles of compounding to locomotives appears to have originated with John Nicholson, a "driver" in the employ of the Eastern Counties Railway Company of England, now a portion of the Great Eastern system, about the year 1850, although there exists some doubt as to this. In 1852 James Samuel, the locomotive superintendent of the same company, read a paper before the Institution of Mechanical Engineers, on "A Continuous Expansion Steam-engine." In this paper, regrettably short and meagre, he gave some particulars of the performance of two compound or continuous expansion locomotives, built apparently in accordance with Nicholson's suggestion.

The cylinders of these engines were placed outside the frames, in a similar position to those of the American engine of to-day, and were 15 and 17½ inches in diameter respectively, the ratio thus being 1 : 1.360. The method of working was to supply boiler steam to the smaller or high-pressure cylinder, and to allow a portion of it to be expanded into the larger or low-pressure cylinder, the amount being regulated to equalize the power of the two cylinders. After an expansion to the lowest useful extent, the steam in the low-pressure cylinder was allowed to exhaust into the atmosphere; that remaining in the other was then permitted to pass into the blast pipe to urge the fire in the usual way. Means were provided for admitting boiler steam direct to the low-pressure cylinder at starting, to obviate the difficulties which would otherwise arise. These engines showed an economy of 20% in fuel (coke) over the ordinary engines, when the latter were cutting off at one-third stroke, the work performed being equal in each case. In competition with ordinary engines using what Mr. Samuel called "the usual degrees of expansion," the compounds gave an economy varying from 54 to 85%, and in some instances even higher. But it must be remembered that in those days the benefits accruing from expansion, although known, were not fully realized, and it was no uncommon thing for the valves controlling the admission of steam to the cylinders to be almost line and line outside, giving a cut-off of about $\frac{9}{10}$ stroke.

In the course of his most interesting paper the author laid down the following important proposition: "The greatest useful effect is obtained from the steam when it is allowed to expand in the

cylinder until its pressure upon the piston just balances all the useless resistances of the friction of the engine itself, and the resisting pressure on the back of the piston (whether the pressure of the atmosphere in a high-pressure engine, or the uncondensed vapor in a condensing engine), the surplus power beyond these useless resistances being alone available for the purposes to which the engine is applied." Although this may sound like the veriest truism at the present day, its full significance is possibly not universally recognized.

What disposition was finally made of these two engines, and why their type was not perpetuated, the writer has been unable to ascertain. Besides being remarkable as the first compound locomotives ever built, they are notable from the fact that the essential features of their design, with the exception of the curious provision made for the blast, and the economy in fuel claimed for them, are almost identical with those of one of the systems now in vogue.

The next advocate of compound locomotives was the French engineer Morandière, who in 1866 proposed an engine having three cylinders, one high pressure and two low. His proposal, however, never assumed a tangible shape, and it is to this country that the honor belongs of actually constructing the first compound locomotive since Nicholson's time. About 1870 the Remingtons, at their works at Ilion, N. Y., built a compound steam-car for the Worcester & Shrewsbury Railroad Company, for suburban traffic. Two cylinders, one 5 x 12 inches and the other 8 x 12 inches (ratio, 1 : 2.5) were used, with means of supplying boiler steam to the larger cylinder as occasion required. This engine gave good satisfaction until its withdrawal from service a few years ago.

In 1873 the late Mr. William S. Hudson, the superintendent of the Rogers Locomotive Works, was granted a patent for a compound locomotive, a feature of which was a superheater placed in the smoke-box. Two cylinders were to be used, the high-pressure "being only about three-fourths the diameter of the other." Like Morandière's design, that of Mr. Hudson was never put in practice.

In 1876 Mr. Anatole Mallet designed and had built, at the Creusôt Works, in France, the three famous compounds for the Bayonne & Biarritz Railway—engines whose marked success unquestionably led to the more or less general introduction of compound locomotives which has taken place within the last ten or twelve years. The history of this type of engine, from 1876 until now, is so well known that further notice here would be super-

fluous.* Suffice to say, that two-, three-, and four-cylinder compounds, to the number of several hundreds, are now at work in various parts of the world, mostly in England, France, and Germany, the resulting economy ranging between 10% and 25%. Two compound engines, one with two and the other with four cylinders, of American design and build, are now under trial in this country, but as they have been in operation but a short time no reliable data as to their performance have been collected.

Within limits, imposed by practical considerations, increase of boiler, or, more properly, initial pressure adds to the efficiency of the steam, as the annexed diagram (Fig. 155), starting with 20 lbs. absolute and terminating with 300 lbs., shows. In plotting this curve, the steam was taken as being expanded down to atmospheric pressure, a condition highly undesirable for a non-condensing engine, but not affecting the relative efficiencies. For a specific illustration, take the difference in the efficiencies of steam at 140 lbs. and

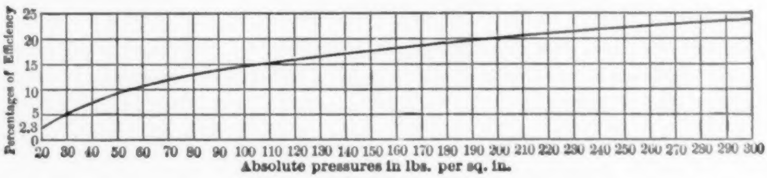


FIG. 155.

200 lbs. gauge-pressure (155 and 215 lbs. absolute, respectively). The temperature of the former is 361° Fahr., and that of the latter 388° Fahr., a difference of 27°. The extra quantity of fuel required to be burned to bring about this increase of temperature is small, 2%† of the amount required for the generation of the 361 steam being over, rather than under, the mark. The smallness of this percentage is due to the fact that the latent heat of the steam at the two pressures is practically the same; consequently the increments of heat go almost entirely to raising the temperature. If the steam in each case was expanded to 25 lbs. absolute, the efficiency of the 140 lbs. steam (assuming expansion

* During this period, the Dunbar compound, having four cylinders arranged tandem, was run for a short time on the Boston & Albany. Its performance did not justify its existence, and it was changed into an ordinary engine.

† The exact determination of this quantity is impossible, for obvious reasons, although the requisite number of heat-units is known.

to take place in accordance with Boyle's law, $P\ V=$ constant), calculated by the well-known formula,

$$E=\frac{T-t}{T},$$

would be

$$\frac{(461+361)-(461+240{\cdot}1)}{(461+361)}=14{\cdot}7\%;$$

and that of the 200 lbs. steam would be

$$\frac{(461+388)-(461+240{\cdot}1)}{(461+388)}=17{\cdot}4\%,$$

an increase of 15·5%.

Correcting for the additional fuel consumed, the actual increase of efficiency would be

$$\frac{100}{102}\times 15{\cdot}5=15{\cdot}1\%.$$

Increase of pressure, however, necessitates greater strength and weight in the component parts of the boiler.

If the thickness of the plates forming the cylindrical portion of a boiler designed for the lower pressure is $\frac{7}{16}$ of an inch, $\frac{5}{8}$-inch material must be used in constructing a boiler for the higher pressure, provided the same diameter of barrel, the same factor of safety, and the same quality of plate and style of riveting are retained. As, however, the efficiency of 200 lbs. steam is greater than that of 140 lbs., the cubical capacity of the boiler may be reduced somewhat; but as the fire-box must be kept of at least the same dimensions as before, in order to prevent an unduly high rate of combustion per unit of grate area, any saving in this direction would be insignificant. Besides the barrel plates, those of the fire-box, together with all stays, braces, etc., must be increased in proportion.

Supposing the 140-lb. boiler to weigh 18,000 lbs., the other would weigh about 25,000 lbs., or 7,000 more. Estimating the price of a locomotive boiler to be 10 cents per pound, this would add $700 to the cost.

Regarded merely as additional weight to be hauled, this increase, amounting to less than half the weight of an empty box-car, would not at all seriously diminish the gain to be derived from the high-pressure steam; but, unfortunately, it must be carried, for the most part, by the already heavily loaded driving-wheels of the engine.

A gauge-pressure of 200 lbs. per square inch would seem to be

about as high as is desirable to go in efforts to increase the efficiency of the steam. Steel plates, which are now used almost exclusively in locomotive work in this country, become unreliable at a temperature of about 450°, and it is, of course, imperative to keep well within the bounds of absolute certainty and safety. Besides this, the objections which obtain in the case of superheated steam, and enumerated farther on, apply with equal force to saturated steam of very high tension.

It may be pertinent to say here that there is no difficulty whatever in obtaining steel plates for boiler work thoroughly uniform and reliable. In the shops in which the writer is employed, with the exception of a few copper fire-box boilers built for South America, nothing but steel has been used for this purpose in five years. Steel for stay bolts, on the other hand, is not an unqualified success; and several locomotive men have gone back to iron as being the better material.

During the last two or three years, in order to avoid the excessive weight per wheel entailed by large and heavy boilers, several railroad companies, running fast passenger trains, have discarded the "American," the generally accepted type of passenger locomotive, for a six-coupled engine of the "Mogul" or ten-wheeled type. Engines of this kind permit of a large adhesive weight while keeping the static load per wheel within reasonable limits; but, owing to the great weight per foot run of driving-wheel base, are more destructive to bridges—except those of very short span—than their predecessors. With respect to increase of internal resistances caused by adding to the number of driving-wheels, it is the firmly rooted belief of many engineers, notably in England, that a "single" engine, having but one pair of drivers, runs more freely at high speeds than one having two pairs of drivers, and so on, owing to the multiplication of bearings and the consequent increase of friction. Nobody, so far as the writer is aware, has ever produced any evidence, apart from generalities, to prove this; but it may well be, and undoubtedly is, true in cases where the road run over abounds in frequent curves of short radius. On lines having a large percentage of tangents, however, the writer has not been able to notice any appreciable difference in freedom of running between engines of the "American" type and ten-wheelers. Although the use of six-coupled engines for passenger service has afforded some substantial relief to the overburdened wheels of the four-coupled machines, the demand for greater power

is always present, and in some recent instances the driving-wheels of passenger "Moguls" and ten-wheelers carry as heavy a weight as the maintenance-of-way engineers will countenance.

Few locomotive engineers would admit the desirability of running 8 coupled engines of the consolidation or 12-wheeled types at high speeds, owing to the great weight per foot run, and if it is really essential that more power be forthcoming, some radical change in locomotive construction is inevitable.

It may safely be stated that no locomotive built during the last twenty years has been deficient in adhesive weight after once getting its train well in motion, notwithstanding the slight and unreliable evidence adduced to the contrary. In other words, the mean effective pressure in the cylinders when running at all fast is quite insufficient to cause the tractive force to exceed the adhesion. A feature of any substantial change in locomotive design will then probably be a method of increasing the coefficient of adhesion at starting, for although many devices tending to this end have been experimented with, we still, in default of something better, have to rely on mere dead weight, a large proportion of which is useless at other times, to get the trains away from the stations.

The evaporative efficiency of a boiler depends upon the nature of the fuel burned under it, and as anthracite and bituminous coal are, generally speaking, alone used in this country for locomotive work, they may be considered to the exclusion of all other kinds. It would naturally be expected that as anthracite is richer in carbon than the average quality of bituminous coal (82 and 58% being the mean of several analyses), it should give a higher evaporative duty. Service trials, however, prove that the difference existing is wholly in favor of bituminous coal, fully bearing out the assertion frequently made by firemen, that a tender-load of soft coal will go further than a like quantity of hard. Recent experiments on the New York, Lake Erie & Western Railroad, with high-class modern locomotives, gave evaporative rates, from and at 212° Fahr. per pound of coal, of 5.68 for anthracite and 7.2 for bituminous. These figures are probably somewhat higher than the average performance of locomotive boilers throughout the country, but represent what is possible and what can be obtained when the firing and handling of the engine are done with reasonable intelligence.

The total evaporative power of anthracite coal containing 82% of carbon is 15¼ lbs. from and at 212° Fahr., while that of bituminous containing 58% carbon is about 12 lbs., due allowance being

made for the other component parts. It is evident that anthracite, as burned in the fire-boxes of locomotives, is an extravagant fuel, whose use is justified only by cheapness of cost, or by some other qualification which is considered indispensable—cleanliness, for example.

Much higher evaporative performances than those mentioned are frequently claimed, as much as 10 lbs. of water per pound of coal in some cases; but although such results are, of course, attained in stationary and marine practice, it is extremely doubtful whether, in locomotive work, they are actually reached. Where such figures are arrived at it is safe to assume, unless the evidence against doing so is overwhelming, that large quantities of water pass over into the cylinders with the steam, while a considerable amount may easily go to waste by way of the safety valves. It is a matter of great difficulty to determine the exact quantity of water fed to a locomotive boiler, as the time allowed for replenishing the tank at way-stations is so short that the swash caused by the rush of water from the stand-pipe does not have a chance to subside before being augmented by the jolting motion of the restarting train. If sufficient time is allowed for taking an accurate reading, the conditions do not conform strictly to regular service, as the time so lost must be made up again by increasing the average speed. The operation of a locomotive boiler militates against a high duty. Its exposure to constantly changing volumes of atmospheric air cannot but be a fruitful source of loss, and the remarkable differences of opinion with regard to boiler proportions, grates, and draught appliances, prove that some boilers, at least, do not have a fair chance to perform their functions in an economical manner.

From the foregoing, the efficiency of a well-designed bituminous coal burning boiler may therefore be taken at

$$\frac{7{\cdot}2 \times 100}{12} = 60\%,$$

which, considering the disadvantages under which it labors, is, the writer thinks, a creditable figure.

DISCUSSION.

Mr. Willis E. Hall.—The subject which Mr. Dixon proposes for discussion is one to which too little attention has been given, and it is strange, but true, that the most extensive information of an

experimental nature concerning the efficient working of locomotives in the hands of the public is that written by D. K. Clark in his *Railway Machinery*, and which was published in 1852-55.

It is a very large field to be covered, and information arising from experiments testing the many points which go to make up the efficiency of the machine should, it is thought, go toward reducing the practice to a standard as near as this can be done.

The compression question is dependent upon the conditions of the service, but in handling it care should be taken not to allow it to be too high, especially with engines which are to run at high speeds, where, with the motion cut back, it is apt to choke the engine and increase the back-pressure and thereby interfere with the useful working of the engine. The percentage of clearance should also be considered in connection with the extent of surface which it exposes for condensation.

Two important elements which enter as economical factors in locomotive performance and which it is desired to mention are:

1st. Steam distribution and working.

2d. Combustion.

The distribution and working of the steam has been brought out prominently recently in the trials and discussions of the application of compounding to locomotives, claiming thereby economy on the ground of a decrease in condensation arising from the less expansion of the steam in each of the two cylinders, although where the high and low pressure expansions of the compound are combined a higher degree is obtained than with a single cylinder. It is simply a case of one and one make two, and the percentage of condensation is dependent, not upon the number of cylinders used in the expansion, but the degree to which the steam is expanded together with that of the amount of surface of contact from which increased radiation will result. That this is greater in compounding is self-evident, but that apparent economy is found in expanding in two or more cylinders instead of one would go to show the loss which has heretofore been attributed to the much-talked-of "condensation and re-evaporation" has been over-estimated. This, too, in addition to the loss which must arise when intermediate receivers are used, whether these be in the form of an independent vessel or so-called steam-pipes leading from the high to low pressure cylinder. Work cannot be obtained from steam without conden-

sation, and it looks as though this had been confused and a larger percentage of it attributed to condensation from radiation and by the walls of the cylinder and passages than properly belonged to it.

The point which Mr. Dixon makes of removing the degree of expansion from the control of the engine-driver is the all-important one and, no doubt, the true secret of the economy which it would appear has resulted from compounding.

Apparent economy is mentioned because, so far as seen, all the trials of compound locomotives have had other influencing causes which would themselves produce a reduction in the coal consumption were they incorporated in the single-cylinder engine. Special reference is made to the increased boiler pressure which influences so much the economy of engine running. To determine the true efficiency of double against single expansion, the design of the boiler and the pressure carried, as well as all the other details of the construction as near as could be, should be the same in both cases. No element of a machine can be tested in any other way.

As regards coal consumption, it is only necessary to call attention to the so-called extended smoke-boxes now so generally used on locomotives to convey some idea of the immense amount of rich combustible material which is collected there to be thrown away at a figure below the original cost of the coal. A reduction in its consumption, whether resulting through higher pressure, compounding, or the use of some form of the so-called vacuum exhaust pipes, or a combination of all three elements, is a question which the near future will develop, and we should not lose sight of the fact that improved combustion is as important a consideration as that of steam distribution and working.

Mr. Angus Sinclair.—The portion of Mr. Dixon's paper relating to cylinder compression in locomotives deals with a subject about which considerable misapprehension exists. When an engineer first examines an indicator diagram taken from a locomotive running fast and cutting off short, he is almost invariably inspired with an ambition to distinguish himself by reforming the locomotive. If he is so placed that the aspirations of theory are unchilled by the demonstrations of experience, he will lose no opportunity to deride the ignorance of the railway engineers who persist in using an engine which produces such a reprehensible card. If the would-be reformer gets the oppor-

tunity to arrange the valve motion of a locomotive to accord with his idea of the eternal fitness of that mechanism, the engine will invariably use more coal in doing certain work than one equipped with the much-maligned link motion.

Most of those who are struggling to improve the distribution of steam in locomotive cylinders are laboring to popularize a delusion. Because certain phenomena would imply very bad practice in a stationary engine, they argue that their occurrence in a locomotive must prove the performance of the engine to be defective. The cases are not parallel. The locomotive of to-day is a machine developed by a tentative process to meet the requirements of peculiarly severe and varying service. The proportions of cylinders and their passages do not vary materially among the locomotives of different railroads or of different countries, and the proportions are undoubtedly those which produce the most satisfactory results in the performance of work. In no instance of general practice has it been found desirable to make the proportions of steam ports approximate to the sizes considered necessary with good stationary engines. When this important feature is so differently proportioned in the two kinds of engines, how can reasonable men expect that the distribution of steam will be nearly alike?

The comparatively small ports and the restricted opening at short points of cut-off make a high compression line absolutely necessary to enable a locomotive to perform fairly heavy work at high speed. If the compression does not reach close to boiler pressure, the steam line will drop so suddenly that the engine can do very little work when the piston speed is close to 1,000 feet per minute. Increasing the size of the card by reducing the compression is a plausible theory, but in practice at very high speed it leaves the main rod to drag the piston away from the end of the cylinder.

The men best able to judge as to what is required to make a locomotive work successfully have decided that high compression is necessary. In many instances, however, compression is carried to excess even for the requirement of a high-speed locomotive. In such cases loss of efficiency and of economy result.

I can scarcely agree with Mr. Dixon's views on steam-jacketing. It is by no means certain that steam-jacketing has proved successful on any high-speed engine; and there are peculiar difficulties encountered in making such a device work properly on

a locomotive. The application of a steam-jacket by Mr. Borodin, referred to in the paper, proved nothing. The locomotive was experimented with in a shed, with small power and running at uniform speed. These conditions are very different from the ordinary work of a locomotive. Few engineers would allow themselves to be influenced by results which are not obtained in road service. I have known of several very careful experiments which were made to ascertain the value of steam-jacketing the cylinders of locomotives, and in every instance the jacket was decided to be of no value.

I agree with Mr. Dixon that it is hard to understand how condensation can take place to any hurtful extent when engines with well-protected cylinders are running at high speeds. I feel compelled, however, to admit that such condensation does take place, and to a much greater extent than 7% of the steam admitted to the cylinders, even though, like Mr. Dixon, I have failed to detect in high-speed cards any trace of condensation or re-evaporation. There is so much of what Hemenway calls "initial expansion" in the cylinders of high-speed locomotives that it is impossible to judge how much of the drop is due to condensation or to wire-drawing. Nozzles and contracted steam-passages exert so much influence on the exhaust end of the card that it is practically impossible to detect the signs of re-evaporation.

Even though that view is taken by such a high authority as Sir Frederick Bramwell, I attach very little importance to the assertions that a vital element in making the compound locomotive a success is the tergiversation of the ignorant engine-driver who refuses to work the steam expansively. If compound locomotives prove successful enough to make that form of engine the railway motive power of the future, that consummation will be produced by the same causes which made marine and stationary compound engines popular with steam users, viz., the capability of using the steam to greater advantage.

Mr. W. O. Webber.—This subject is one in which I have taken a great deal of interest. It has always seemed to me that the locomotive-engine was the most maligned engine on the face of the earth. You always hear how wasteful the locomotive is; how inefficient; how uneconomical. I do not think that is giving the locomotive a fair show. It has always seemed to me that the modern locomotive represented one of the highest types of

the high-speed engine, and under the very worst possible conditions. You have a high-speed engine of very large power under the most unstable conditions—practically no foundation at all, running in a whirlwind of dust, and with every possible condition against the engine; and yet I think it is only fair to say that, taking the average of locomotives running throughout the country to-day, and their performance is wonderfully efficient, considering the circumstances under which they are running. There is a great deal to be said about the management of locomotives. In conducting a great number of trials, primarily taken as coal tests to determine the relative values of different classes of bituminous coal on locomotives, it became necessary to test the locomotives, and also to test the engineers and firemen as well. I desired, of course, in making my tests, to get the best possible results out of the coal being tested and the engine being used, and in order to do this I found that it was very necessary to instruct the firemen how to fire the engine properly; and with the locomotive-engineers very much in the same way. The latter invariably insist upon running an engine with the throttle instead of the reverse lever; and right here I think it might be said that some of our locomotive-building friends ought to pay a little more attention to the reverse lever than they do. Superintendents of motive power of roads complain that engineers do not run their engines with the reverse lever. The trouble is not entirely with the engineer; a good deal of the trouble is with the reverse lever. I have ridden a good many thousand miles on a locomotive, and in very few instances have I seen the train and the grade and all the other conditions so proportioned that the engineer could put the reverse lever, say, in the 8-inch or 10-inch notch with the throttle wide open and keep it in there any length of time without losing time or gaining time. As he strikes a down-grade he has got to pull his lever up or shut off, and *vice versa*. I think the reverse lever needs a good deal of attention. The ordinary running of a locomotive engine, I think, gives remarkably good results. I have got here in my note-book a number of tests which, as I said before, were not made to test locomotives, but rather to test the coal; but I find here evaporations running from $6\frac{1}{2}$ to $8\frac{1}{2}$ and 9 lbs. of water per pound of combustible, and the fuel consumed per square foot of grate surface 90 lbs., and running from there to 136 lbs. These engines were small

engines. One engine on which most of my tests were made was an engine with a fire-box only 3 feet wide and 5 feet long, with a 42-inch boiler, 114. 2-inch flues, very small exhaust nozzle, $2\frac{1}{2}$-inch; engine 15″ by 22,″ and only 740 total square feet of heating surface.

Mr. Geo. S. Strong.—I fully agree with the speaker that the locomotive is not a wasteful machine as long as it is not pushed beyond its capacity. The difficulties which we have to deal with to-day are the handling of trains which call for 1,000 to 1,500 H.P. and to handle them with an engine which can economically develop 600 H.P. The American locomotive, or the ordinary slide-valve engine with the link motion, as we all know, is capable of developing a horse-power to about 27 lbs. of water when it is not overloaded. As our friend has just remarked, the evaporation under ordinary conditions will run from $5\frac{1}{2}$ to 6 lbs.; but when we push it beyond the economical point and undertake to burn more than 150 lbs. of coal per square foot of grate surface, then the evaporation sinks very low and with it the capacity of the engine to do work when the cut-off is beyond 8 inches or 12 inches, and the engine becomes a very wasteful engine. In fact, the coal is lifted from the grate and the steam is not properly expanded. The advantages of compounding, of course, are very largely due to the increased expansion more than the difference in condensation of cylinders. The increased capacity of the engine due to the different steam distribution is very readily understood. The question of compounding or getting our expansion in some other way is matter of choice. We have shown by tests that we get about the same economy that they get out of the compound engine. At the same time, I think there is a chance for improvement still with the 4-valve engine by compounding. I have designed a compound engine in which I regulate the point of cut off in the high-pressure cylinder independent of the low-pressure, and will get the same distribution in the compound locomotive that we get in the compound automatic engine. With that engine I hope to get a horse-power with 18 lbs. instead of 27 lbs., as the ordinary locomotive gives.

Mr. D. L. Barnes.—These statements ought not to be made regarding locomotives without some contradiction. I think it behooves the members to take more notice of them. It is impossible, with economy, to expand in a simple engine as much as in a compound engine, and it is a rather amusing communication

that has just been read stating that on locomotive engines without compression the driving-wheel would have to draw the piston and connecting-rod away from the end of the cylinder, particularly in engines where we are troubled with pre-admission to the extent of about $\frac{3}{32}$ of an inch. If there were no compression there would be sufficient pre-admission to move the piston away from the end of the cylinder. It is also difficult to understand the connection between the compression line on the indicator card and the size of the steam-passages.

With regard to running entirely with reverse levers, it is fortunate for the railroads of the country that the engineers do not run that way, because there are many times when it is not economical to do so. When the locomotive is priming badly, as it often does when steaming hard, the throttle should be partially closed and the steam wire-drawn a little. Almost all locomotives prime more or less at times.

With regard to the water evaporated per pound of coal, I see the statement is made that it is as high as 9 lbs. This statement probably results from inaccurate observation. A large portion of the water supposed to be evaporated passes into the cylinder in the form of water.

Regarding the weight of water used per horse-power, I know of no experiment which accurately records the amount of water in the shape of steam used per horse-power on the locomotive. The measurements taken from the tank and from the indicator cards are all that is known regarding the water used per horse-power. These indications are not sufficiently complete to enable the weight of steam used to be determined.

In this paper I notice the statement that "the writer has examined a large number of cards taken from locomotives at speeds ranging from 40 to 65 miles per hour, but has been unable to detect any trace of either condensation or re-evaporation. Supposing, though, that some small amount of steam, say 7%, was condensed at each stroke, why expanding in two cylinders, one presenting twice the area of the other, should diminish it is almost beyond comprehension." I would like to have any one who is familiar with locomotive indicator cards at that speed determine even where the point of cut-off is, to say nothing about the cylinder condensation.

Regarding compounding locomotives at slow speeds, the value of the statement does not appear. If compounding is good at

slow speeds it is good at high speeds—perhaps not equally so, however. The reasons, I think, are obvious to all. I have not found, from examination of all the engine tests that I have been able to procure, that running with increased speed above ordinary speed produces a decrease in cylinder condensation. But the statement is here made.

Regarding the effect mentioned in this paper of Moguls and American locomotives on bridges, I think the statement is not true. A Mogul locomotive is easier on a bridge than an American locomotive, for two or three reasons. The longer the bridge the less the concentration of wheel loads affect the bridge as a whole. Again, in Mogul engines the weight of reciprocating parts balanced in the wheels is divided among the wheels, and as these weights in eight-wheel engines produce probably the most detrimental action on the individual members of the bridge, it would appear that Mogul locomotives are easier on bridges.

Mr. Louis S. Wright.—It has occurred to me, while listening to these arguments, that a number of comparative tests of locomotive efficiency might be made in the shop by standing a locomotive on a frame, as shown in Fig. 222, and operating it on wheels in place of rails, anchoring it fore and aft with a spring-balance, or some measuring device, so as to register pulling force. This would reverse the usual method of testing locomotives working under load, by making the locomotive stationary and having rails—which in this case are wheels—to move. The working load is made by applying friction brake to the enlarged portion of the axles of the wheels in the frame, as shown in the sketch. The journals of the axles of the wheels in the frame are hung on levers, with a spring-balance at the end of each lever. This will register the weight of the engine on the driving-wheels, and the variation of the spring-balance on the levers will register the force of the hammer-blow struck at the various speeds of the locomotive when running. The efficiency of the engine on grade can be measured by inclining the frame on which the engine stands to any grade required by pivoting the frame at one end and lowering the other end.

Indicator cards, which are hard to obtain at the high speeds of locomotive in service, can here easily be made. Amount of water and coal consumed can be measured with ease. Of course, all tests made in this way would only be comparative,

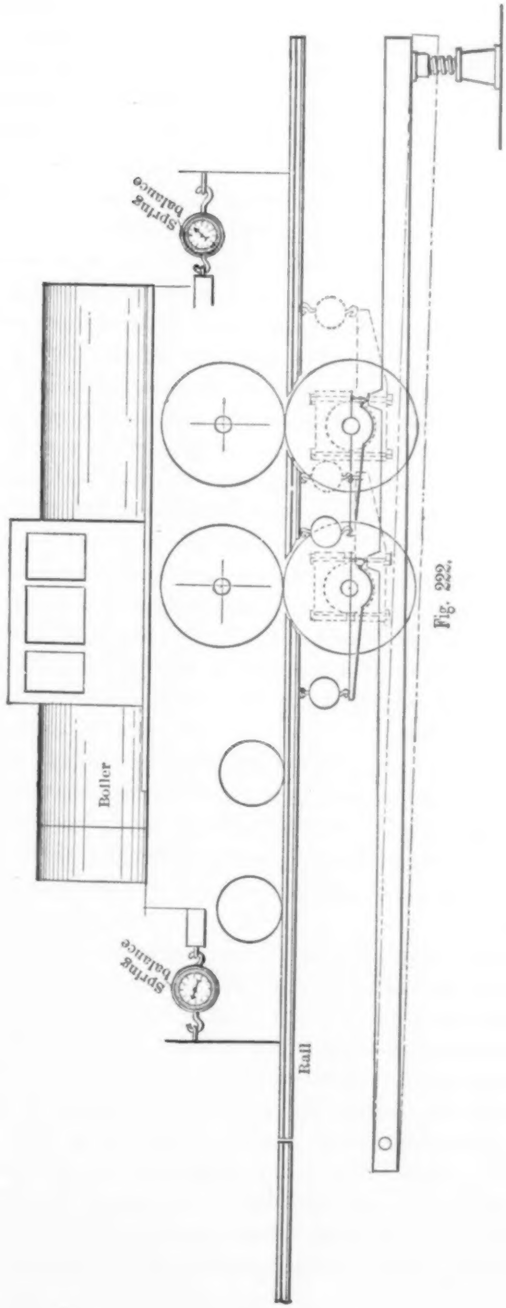


Fig. 222.

and would probably not agree with tests made in actual practice on the road, but at the same time they might give some very reliable data. I do not know that anything of this kind has ever been attempted, but as the suggestion has offered itself to me, I thought possibly it might be a means of bringing about some desired ends, viz., easily-made, reliable tests.

Mr. W. F. Durfee.—Following out the thought which has just been presented, I will say that in 1852, on the borders of Fresh Pond, near Cambridge, Mass., there was an arrangement of precisely the character suggested used for hoisting ice. The locomotive was run in from the track upon two pairs of friction wheels beneath the rails, so placed that the drivers could be run upon their tops without touching the rail. The engine was anchored with heavy turn-buckle screws to posts, and the power was taken from the axes of the friction wheels for the work of the ice-house. When they had finished gathering the ice crop the locomotive was run back upon the road for use in ordinary service.

Mr. L. S. Randolph.—In regard to the efficiency of the locomotive, the question has been and is now not so much what is the efficiency of the locomotive, but what is her power? Not so much how many foot-pounds of energy it will develop per 100 lbs. of combustible, but how many cars it will haul and make schedule time.

It would seem, from present information on this subject, that the above view of the case is the correct one; for within certain limits the efficiency of the railroad as a machine for transportation is of more importance than the efficiency of the locomotive, and, in fact, the latter obtains its value only from its effect on the former.

The tests of the efficiency of a locomotive as usually made are not only exceedingly difficult, but not very reliable. The varying factors of grades, curves, and speed, together with the comparatively short time it is possible to give to the tests (not more than 5 or 6 hours at the most), make it impossible to get the accuracy and reliability which the subject demands. Not until we take a leaf from marine engineering and make, as it were, "dock trials" of a locomotive will we get the necessary degree of accuracy and reliability.

With the locomotive blocked up, a pillow block substituted for the driving box, and a prony brake to measure the power

developed, we should be able to get with accuracy the efficiency of the locomotive under varying conditions of load, speed, etc., other conditions being kept uniform. Knowing these points accurately, we could determine how much was gained or lost by over-loading or under-loading an engine, as well as the effect of varying dimensions.

Mr. A. T. Woods.—There are some other points in this paper which I think should not be allowed to pass without comment. Mr. Dixon intimates that while the benefit of compounding is beyond dispute at slow speeds, it is of doubtful advantage for express locomotives. As a matter of fact, there are triple and quadruple expansion marine engines running at piston speeds of from 800 to 925 feet per minute. I do not see that we can draw the line and say that such speeds are slow for engines having three and four cylinders, and that 1,000 or 1,100 feet per minute is fast for engines—such as compound locomotives—in which the steam is used in but two cylinders in succession.

Referring to Fig. 154, Mr. Dixon states that in a certain case it was found necessary to compress to 54 lbs. with 10% clearance, and he then makes a comparison of mean pressures on the basis of compressing to the same final pressure with but 2% clearance. The work of compression is very different in the two cases, and the comparison is therefore not correct.

It is not clear to me that the fact of the crank-pins being worn, as the author states, is substantial proof of excessive compression, since the same result might follow from the variation in pressure due to an early cut-off even without compression.

Mr. Wm. Kent.—I wish to call attention to an error in Mr. Dixon's paper, where, on the twelfth page, the evaporative power of anthracite containing 82% carbon is put at $15\frac{1}{4}$ lbs. As the evaporative power of carbon is only 15 lbs. $= \frac{14500}{966}$ heat units, and the volatile matter in anthracite is generally of no heating value, the O balancing the H, 82% coal cannot be considered to have more than $.82 \times 15 = 12.30$ lbs. evaporative power. Bituminous coal, on the contrary, contains volatile matter of great heating value. Cumberland semi-bituminous is both theoretically and practically better than anthracite when burned in a good furnace (see my paper on evaporative power of bituminous coals, in Transactions, Cleveland meeting). Bairns' numerous tests will no doubt also confirm this. I have just tested a semi-bituminous coal which gave only 8.42% ash and refuse coal in the boiler

test; it will show probably not over 3% ash by analysis. No anthracite is equal to this.

I would also call attention to error in the method of stating percentages in Mr. Carpenter's paper on tests of engines (see p. 31). Variation in speed .031% should be 3.1% or .031 without the per cent. This error is repeated through the whole paper. Mr. C.'s calorimeter tests are exceedingly improbable. That two consecutive tests (see p. 32) should show 0 and 9% of moisture, and that an ordinary tubular boiler (pp. 20 and 22) should prime 11% (unless its water-level were at the top of the boiler) is incredible. I think Mr. C. is also in error on p. 9 in allowing 2 H.P. for the injector, adding it to the engine. An injector taking steam out of a boiler and putting the heat of this steam into the boiler again in the water it feeds on "consumes" no heat. All the heat energy which goes into the injector reappears in the boiler again, less that lost by radiation and leakage. (See my discussion on Prof. Webb's paper on this subject a year or more ago*.)

I think a similar error is made by the Engine Test Committee (p. 18 of their paper, second paragraph, "Where an injector is used a deduction is to be made for the increased temperature of the water derived from the steam which it consumes"). When I make a boiler test with an injector I take the temperature of the feed before it enters the injector, never after it leaves, and make no calculation of any kind in regard to the injector.

Mr. W. F. Dixon.†—Objection has been raised to the statement that although the benefit of compounding is beyond dispute at slow speeds, it is of doubtful advantage for express locomotives, on the ground that there are triple and quadruple expansion marine engines running at piston speeds almost equal to those of high-speed locomotives.

I am of the opinion that piston speed *per se* has little to do with the question of cylinder condensation, but that the number of strokes per unit of time is the important factor. The length of time per stroke during which any specified point of the surface of the cylinder is exposed to the chilling in the end of the exhaust is much shorter in the case of the locomotive with its 2-foot stroke than with the marine engine with its 5 feet or thereabouts.

As to the accuracy of the statement that anthracite contain-

* Trans. A. S. M. E., Vol. X., p. 341.

† Author's Closure, under the Rules.

ing 82% carbon can evaporate 15¼ lbs. of water from and at 212° Fahr., I may say that an analysis of the fuel in question showed it to contain 7.4% of volatile matter of an evaporative value of a little less than 3 lbs. of water at 212°. This added to the $.82 \times 15 = 12.3$ lbs. evaporative value of the carbon alone gives 15¼ lbs., as stated. Although made clear in the paper, it may be well to add that this figure is not intended to be mathematically but only approximately correct.

CCCXCII.

AN OPEN MERCURY COLUMN FOR HIGH PRESSURES.

BY W. W. BIRD, WORCESTER, MASS.

(Junior Member of the Society.)

THE mercury column described in this paper is the result of an attempt to provide for the testing of spring gauges used in connection with the work carried on in the Mechanical Laboratory of the Worcester Polytechnic Institute.

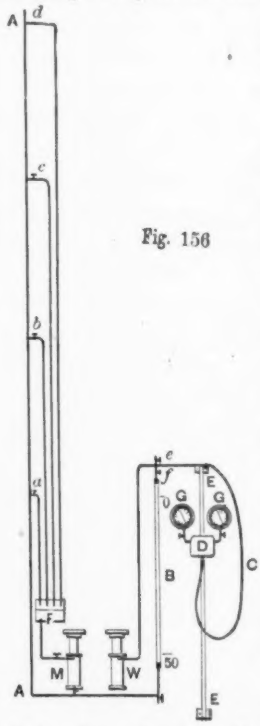


Fig. 156

The object was to design a column which would give pressures from atmospheric to 200 lbs. to the square inch, or higher if needed; it should be sufficiently accurate for engineering purposes and at the same time not expensive, and, if possible, it should provide for taking readings at one convenient point.

Referring to Fig. 156, *AA* is a small iron tube connected at the lower end with the glass tube *B*, which in turn is connected by a flexible tube, *C*, with the gauge or gauges *GG*. They are fastened to the carriage *D*, which runs on the rod *EE*. The tube *AA* in this case is made up of four sections, each section being of sufficient length to give a pressure of 50 lbs. when filled with mercury at 62° Fahr.

The method of operating is as follows: By means of the pump *M* mercury is forced into the column. It rises in both the iron and glass tubes, the cock *e* being open. When the mercury in the iron tube reaches the end of the first section it passes through the valve *a* and down an overflow pipe to the collector *F*. This fixes the zero mark on the scale. Water is then put in at the top of the glass tube

by means of the pump *W*, and fills the top of the glass tube and the flexible tube *C*. Closing the cock *e*, more water is forced in, and the mercury in the glass tube is displaced, thus forcing more into the iron tube, the result being an overflow at the valve *a*. Thus the mercury can be lowered to any desired reading between 0 and 50, while the level in the iron tube remains the same, the gauges being moved up or down to correct for the column of water. The operator at any time can easily assure himself of the level of the mercury in the iron tube by forcing in more mercury until some comes down the overflow.

When pressures from 50 to 100 lbs. are desired, the gauge is corrected or its correction noted, if any, for 50 lbs., by the first section, the valve *a* closed and mercury forced in until it comes down the overflow from the valve *b* and stands at 0 in the glass tube, the water having been returned to the pump or allowed to escape. The gauge being brought to the proper place, the reading should be 50, or show the same as at the end of the first section. This gives a means of checking the measurement of the tube from *a* to *b*. The operation is now the same from 50 to 100 as from 0 to 50. And by closing the valves *b* and *c*, pressures from 100 to 150 and 150 to 200 are obtained. More sections can be added if higher pressures are desired.

By closing the valve *f* two gauges can be compared, by changing the water pressure with the pump *W*.

By some simple means the valves *a*, *b*, and *c* could be opened and closed from below, and thus a gauge could be tested from 0 to 200 or higher, with an open mercury column, without going up more than eight or ten feet to take readings.

The column at the institute was built in general after this plan and is giving good results.

DISCUSSION.

Prof. D. S. Jacobus.—In connection with the mercury column described by Mr. Bird, I wish to illustrate one which is now in process of construction at the Stevens Institute of Technology. A column involving the same feature as the one I wish to describe was designed by Francis B. Stevens; this has been in use for many years and given entire satisfaction. The column at the institute is designed as follows: *E* and *D* (Fig. 227) are two wrought-iron pipes of about ½ inch internal diameter, containing

mercury. *A* is an air reservoir from which compressed air is led through the pipe *B* to the upper end of the pipe *D*. When the air in *A* is not compressed above the pressure of the atmosphere the mercury in the pipes *E* and *D* is at the height indicated by two lines marked on the small glass disks *N* and *O*. *J* is an iron float resting on the mercury in the pipe *E*. A silk string, *G*,

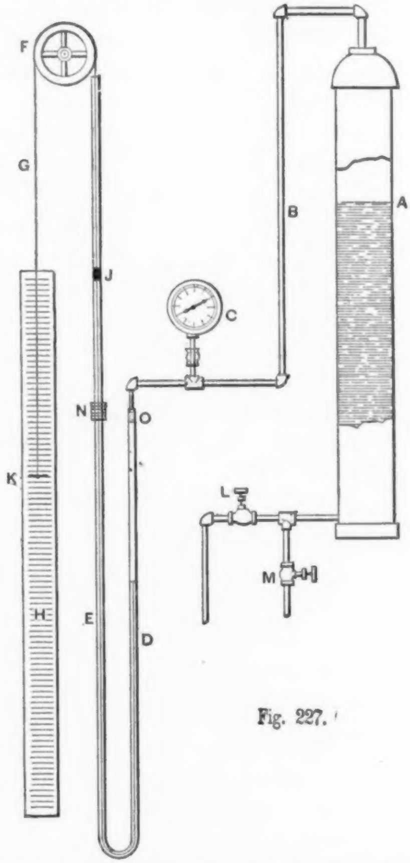


Fig. 227.

crosses from this weight over the pulley *F* and supports the index *K*, which slides along the scale *H*. *L* is a valve for admitting water to the reservoir *A*, and *M* a valve for releasing the same. The method of operating the column is as follows: The gauge to be tested is placed at *C*. Water is admitted to the reservoir *A*, thereby causing compression of the air above it. This compressed air acts on the gauge and on the top of the mer-

cury in *D*, thereby causing the mercury in *D* to fall and that in the tube *E* to rise. The amount that the mercury is elevated in *E* is shown by the position of the index *K*. The weight of the air in the tube *D* may be neglected for all ordinary pressures. In the instrument designed by Mr. Stevens the tubes *E* and *D* were made of precisely the same bore, so that the amount that the mercury was depressed in *D* would be the same as the amount that it was elevated in *E*. In the column at the institute this refinement has not been gone into, for which reason the following method will be employed in graduating the scale *H*. A gauge is taken to one of the standard makers, and compared with a mercury column used by them. This gauge is carried back in a careful manner, placed at *C*, and employed in obtaining the graduations on the scale *H*. After graduating *H* the gauge is again compared with the standard mercury column. If it still agrees with the standard, it may be assumed that it was correct at the time that the scale *H* was graduated. After this second comparison with the standard the gauge is again placed at *C*, and the graduation of *H* verified. We thus obtain a column that, when once standardized, may be used for testing other gauges. The arrangement of the silk thread leading over the pulley *F* has been found to give entire satisfaction in the case of several mercury columns now in use, so that there need be no apprehension that it will give trouble by stretching. The advantage which this gauge has over many others employing mercury is the ease with which it may be used, there being no complicated adjustments or corrections required. The scale *H* is only one-half the length of one which must be employed in a column having a single mercury tube, and for this reason it is much more easy to provide means for reading it along its entire length.

CCCXCIII.

HEATING FURNACES.

BY D. K. NICHOLSON, STEELTON, PA.
(Member of the Society.)

A SIEMENS regenerative gas furnace will probably call for less repairs than any other kind of a reverberatory furnace. The gas and air, coming up separate ports, do not unite until they have reached the hearth, where they come in contact with the metal to be heated. By the time the flame reaches the outgoing ports it is pretty well spent, and does very little more injury to the brickwork than the incoming gas and air. The bricks are not subjected to such a cutting action of the flames as they are in a coal furnace.

The heated regenerators give the gas and air an upward tendency. This makes an outward pressure when the chimney damper is partially closed. It is necessary to work the furnace with an outward pressure when it is charged close up to the door, to fill the furnace with flame and keep the cold air from drawing in at the doors, or when the charge on the hearth of the furnace is up to the required heat and the drawing of it is delayed for a short time.

A Siemens furnace of a simpler form, where the ports are done away with, is known as the Smith furnace. The regenerators are built in the ends of the furnace and are entirely open at the top, which is on a level with the bridge. The bridge of the furnace at the same time serves as one wall of the gas chamber. This furnace is generally used for quick heating. Its advantages are the cheapness, the ease with which the checkers can be taken out and cleaned, and the bottoms of the gas and air conduits and the checker chambers are so little below the surface that they are not likely to become choked by water in low ground. Possibly the gas will not come in the furnace with as much pressure as in an ordinary Siemens furnace, unless the gas-pipe from the top of the gas-producer

stack is correspondingly higher. Gas is purer when it has a long way to travel from the producer to the furnace: the soot has a chance to deposit on its way, but the expansion and contraction of a long overhead tube must also be considered and provided for. The gas checkers often become clogged with this soot. The best way of cleaning them is through spontaneous combustion, which is brought about by shutting off the gas entirely from the furnace, placing the reversing valves on the centre, and opening the stack damper, and at the same time a door of the furnace. There will then be a free passage of air from both ends of the hearth down through the checkers. A small amount of reddish matter will be left as a product of the combustion. When a gas made of vaporized oil and steam is used as the fuel, it comes into the checkers with such force that it will find its way through a choked mass of checker-work, when coal gas, with its gentle pressure, will not.

The hearth of a furnace is not of equal temperature over all its surface. There are different reasons for this: the design of the bridge, the arch in the roof tending to draw the bulk of the fire through the middle of the furnace, and the cold air at the doors chills off the front. If the piece of metal reaches all the way across the furnace, and the gas and air enter side by side or mix before coming over the bridge, there is very little difficulty in putting an even heat on the piece from end to end.

To do good and careful heating the bottom of the furnace must have plenty of attention, especially when the stuff is charged by a mechanical device. The scraping over the bottom soon tears it away. With good fire-sand and care there is no trouble in maintaining a clean, hard bottom, gradually sloping toward a tap-hole at the back of the furnace, midway from the end. This, besides being a help to the heating, makes charging and drawing easy. The bottom can be made and maintained with a slope toward one tapping-hole easier than to two or even three. A high place on the hearth can be cut down by putting some fine coal on it; the ash of the coal combines with and fluxes the silica of the sand.

The process of "burning down" a high bottom is injurious to the furnace, since it is necessary to raise it to such a high temperature that the bricks in the roof drip.

A sand with too much clay in it is liable to cling to a piece of

steel all through the rolls, and show in yellow streaks on the finished bar.

When the cinder sinks into the blow-holes and unevenness of the steel, and then is lengthened into streaks in the working of it, the finished stuff is termed snaked. These snakes show to the prejudice of the steel when it is highly polished, as for instance in various weapons. A seam is where the metal is actually parted ; it is generally called "piping." A low-carbon steel stands nearly as much heat as iron. In heating high-carbon steel the metal must be very gradually brought up to a heat sufficiently high for working it. To charge cold steel of high carbon into a warm furnace is disastrous to the metal, particularly when the steel is frosty. It is an easy matter to build a wood fire around the steel and warm it throughout to at least

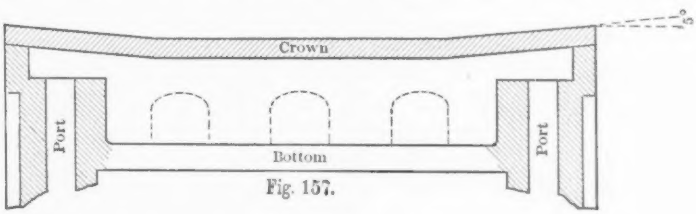


Fig. 157.

200° or 300° before charging. This will mitigate the stress on the inside when the outside begins to expand with the heat of the furnace.

A very fine steel, not too high in carbon, will scale off in sheets, while the scale from a high-carbon steel will break up into fine powder as it comes off. This can be more readily seen under the hammer than at the rolls, where there is generally so much water. Gas which contains steam oxidizes the steel and makes a good deal heavier scale than other gases, ordinarily from 1.25 to 2% on ingots charged hot.

Furnace crowns have been built on a great many different plans, but, after all, the one that gives the most satisfaction is a roof that dips about five degrees at each end for one-quarter the whole distance, and then straight across, with the necessary arch sideways for support (Fig. 157) ; or even a straight roof, with the necessary arch. A very rough crown obstructs the draught across the furnace; the unevennesses create eddies.

The height of the crown above the hearth depends somewhat

on the size of the pieces to be heated, and for cold metal generally four or five times the thickness of the pieces. When the metal is only re-heated, it is not necessary to have the roof so high.

By exposing a piece of steel to too high a temperature in the furnace, a crystalline structure is induced which renders the metal unfit for further hammering or rolling. The evil effect of this overheating is, partially at least, remedied by lowering the temperature to a point below that at which it is generally worked and gradually bringing it up again to the right degree. This is attained by various devices, the most common of which is to pass a highly reducing flame over the hearth. When the steel is overheated or burnt to a greater degree, the burnt part is covered with a flat piece of wood, or some sand, a short time before drawing; and sometimes coke or charcoal dust is spread over the bottom of the furnace, and the overheated parts turned into it. Besides bringing about a lower temperature, all these methods shield the metal from oxidation. It is possible to restore steel and iron almost to what they were before being overheated by working the piece a great deal. At its best, however, the metal is treacherous, and on any account it should not be used where much depends on its strength.

The top and sharp corners on the top side of a piece of steel are the only parts liable to become overheated. The bottom of the furnace being in contact with the under side of the piece, keeps it lower in temperature than the sides and top, so much so that it is necessary to turn the piece, that the under side may be on top a short while before drawing, in order that it may have an even heat. For these reasons pieces of steel, at the stages in their working where they have to be re-heated, should be left in such a shape as to favor these things. The round corners on ingots and the fillets in the grooves of blooming rolls are in favor of both the heating and turning. A very large slab cannot be turned on the hearth in any convenient manner. In this case one end is propped up with a fire-brick, to give the flames a chance to sweep under it.

In a coal furnace for heating iron or mild steel the hearth has a gradual slope from the bridge to the velvetry, which is the hottest part of the furnace, and is tapped at the base of the chimney, giving advantage of all the bottom. For working steel the bottom is sloped toward a tapping-hole at the back, near the

velvetry. The furnace is not worked hot enough to be tapped at the flue.

Various plans are put into practice to aid in burning the smoke and gases driven off a green fire. These are, however, generally abandoned after awhile, and the furnace is worked with simply a fire-box and forced draught, or a combination of natural and forced draught. In introducing these schemes for burning the gases, very often the advantages from them are over-estimated, and the grate surface is made too small. Coal furnaces are now built when it is not considered worth while to go to the expense of building gas-producers. They are convenient at a hammer where a high heat is wanted for a short time; but, generally speaking, they have had their day. A coal furnace will not compare with a gas furnace, where an almost even temperature is held from week to week, especially when it is only required to bring metal which is already warm up to a hammering or rolling heat.

Explosions are prevented, in starting up a gas furnace, by building a fire on the hearth to heat the regenerators and surely ignite the gas as soon as it flows out of the port and mixes with the air. The principal virtue of the heated regenerators is in their creating a draught. The chambers are filled with the gases from the combustion of the fuel on the hearth and air if an excess has been passing over, so that when the gas is let on the heated chamber causes it to ascend so rapidly that there is very little danger of its forming an explosive mixture with whatever air there may have been in the chamber. By introducing little trap-doors at each end of the gas valve-box, cold air may be admitted, with the outgoing current, to cool the gas valve. This checks the draught through the gas checkers to a certain extent, and sends an extra amount through the air. These arrangements are used in melting furnaces, and will hardly be found necessary in heating furnaces. Natural gas is not passed through regenerators, but introduced at the end of the furnace. There is no danger in turning it on when there is a fire on the hearth.

Vertical or pit furnaces are only used to reheat ingots brought to them soon after being cast. These furnaces save labor in charging and drawing, but they are not equal to horizontal furnaces for heating an ingot evenly. The greatest stumbling-block to be contended with in their working is the getting rid of the cinder, especially when low-carbon steel is being heated.

This is attempted in two ways: one in tapping through the bottom at the centre of each compartment, and the other in sloping the bottom of all the compartments toward one end, where there is a tap-hole, the same as in an ordinary furnace. It is a difficult matter to keep this gradual slant on account of the tendency of the bottom to pile up under the partitions that divide the furnace into compartments. Tapping through the bottom is the safer and better way of these two, if the tap-hole could be easily opened and closed. This draining off of the cinder is a most serious matter, as it so easily cuts its way down under the ports to the crown of the nearest regenerator chamber. Altogether the best design seems to be a pit with only one partition and a tap-

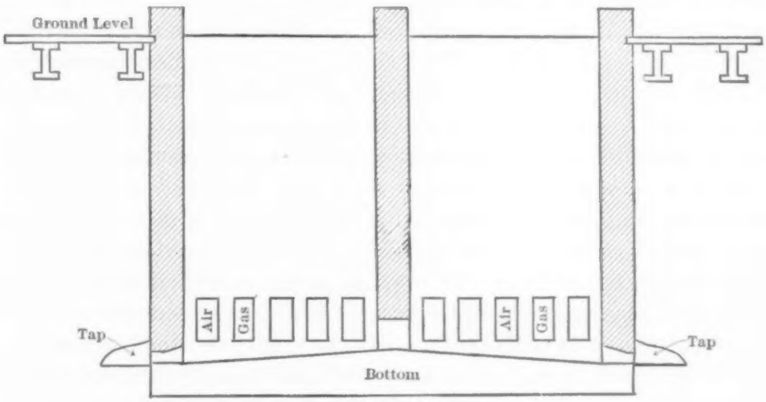


Fig. 158.

hole at each end of the hearth (Fig. 158), with the regenerator chambers removed a safe distance from the bottom of the furnace. Although the ports are generally only a few inches above the bottom, and a sharp flame is liable to burn the butt-end of the ingot, there is no advantage in raising the ports. In fact, they do not heat so well when the ports are raised much above the bottom. These furnaces require, from different causes, a good deal of repair. Considering this and the disadvantage of having everything underground, it is a question whether they are more economical than horizontal furnaces. It might be added that it is possible to heat cold ingots in these pits, but it is not at all profitable.

Soaking pits, of course, require no fuel, and the ingot is handled with more speed and ease than even in the pit furnaces,

because only one ingot is put in a pit. In a soaking pit the ingot must be charged very soon after it is cast, so that there will be heat over and above what is necessary to give a uniform heat to the ingot, to furnish heat for that which is lost from the sides of the pit. It is possible to charge the ingots too soon after being cast; for instance, where they can be swung around and put in the soaking pits by the same crane that lifts them from the casting pit. When the Bessemer is running very fast, there would hardly be room enough to let the moulds and ingots stand in the pit until the time when the ingots would be fit to charge. This can be avoided by casting in moulds mounted on a truck, and then taking the whole affair away from the Bessemer to the soaking pits, within a reasonable distance (50 to 75 yards). While keeping the ingot on its end from the time it is cast till the time it is put on the table to be rolled throws all the advantages on the side of the ingot in cooling down from the liquid to the solid state on the inside, by the nature of the case the stuff is not likely to roll so well from a soaking pit as that which is heated in a furnace with fuel supplied to it. The inside is bound to be softer than the outside and more yielding, and throws more stress on the outside, while being hammered or rolled, than is its share. The stress on the outside is not backed up by resistance on the inside, and so a weak place will, more readily, give way and crack, while in a furnace the outside is more likely to be as hot as the inside, or even hotter, since the heat comes from the outside as well as the inside. In ordinary running at least 90% of the ingots from the Bessemer could be cared for by soaking pits. This is a considerable saving of fuel, labor, first cost, and practically no repairs.

CCXCIV.

EQUILIBRIUM ARCH CURVES.

BY HENRY HARRISON SUPLEE, PHILADELPHIA, PA.
(Member of the Society.)

NOTWITHSTANDING the great antiquity of the use of the arch in construction, the true theory of the equilibrium of forces in its figure was not propounded until the latter part of the seventeenth century, when, after the solution of the problem of the catenary by the brothers James and John Bernoulli, Dr. David Gregory announced, in *Philosophical Transactions* for 1697, that "None but the catenary is the figure of a true legitimate arch, and when an arch of any other figure is supported it must be because in its thickness some catenary is included."

Although two hundred years have elapsed since this principle was thus definitely announced, yet it is well known that arches are not built upon catenarian lines in actual practice, and that the usual manner of designing arched structures, whether of masonry or in framework, is to select a curve, usually composed of circular arcs, generally chosen with a view of producing a pleasing effect, and then to determine, with more or less exactness, the position of the equilibrium curve, stability being secured by making the arch ring of sufficient depth to insure the retention of the equilibrium curve well within its limits. Many arches, indeed, are built upon purely empirical methods based upon previous examples, and although the mathematical treatment of the subject has been very fully discussed by able writers, yet the lack of a convenient working method based upon correct principles appears to be the principal reason for the use of such empirical methods.

The common catenary is the curve assumed by a suspended chain bearing a uniform load, and this by inversion would give the correct arch curve for a similar load, all tensions being converted into compression of equal magnitude and reversed direction. If this were all, the problem would be a simple one; but

in nearly every case the load upon an arch is not uniform, but is distributed unequally at various points of the span. This produces what Rankine terms the transformed catenary, and in his treatise on civil engineering he gives a method of determining the ordinates for points in such a curve for a given load line.

The method is far from simple, however, and is not general in its application, and is hardly simple enough to be of much use in general practice.

One of the most satisfactory discussions of the equilibrium of the arch for customary forms is that of Professor Fleeming Jenkin in the article on bridges in the ninth edition of the *Encyclopædia Britannica*, and yet in summing up his discussion he remarks: "Practically the thickness of the arch ring is determined by rules derived from experience, and the chief use of the above theory is to determine the dimensions of the abutments."

It is proposed in this paper to discuss a simple method, based upon correct theoretical principles, which will give, in a simple and direct manner, the proper equilibrium curve for any arch of given span, rise and load, with a degree of accuracy which is such a close approximation to rigid exactness as to be entirely within the limits of the errors of construction.

A brief reference to the principles involved in the use of the catenary may be permitted to introduce the method.

When a chain or flexible cord is suspended by the ends and permitted to hang freely in any curve which it may assume, it will be found that the shape of the curve will depend entirely upon the amount and position of the load which may be suspended from it. If the chain simply carry its own weight, the load is uniform at all points, and the curve is that known as the common catenary—very closely approximating to a parabola; while if the load be unequal, as caused by suspending various weights at different points, the curvature will be found to be dependent upon the position and magnitude of the various weights, and any change in any weight will be followed by an entire readjustment of the curve throughout its whole length. The curve is thus at all times in equilibrium, whether equally or unequally loaded, a continual balance of forces existing.

Two properties of the catenary enable us to determine points in the curve for any given loading, when considered as an arch: (1) The horizontal component of the tension in the curve is the

same at all points. (2) The vertical component of the tension in the curve at any point consists of the weight at that point, together with the sum of all the weights below that point.

It is only necessary to invert the catenary and reverse the strains, converting tension into thrust and vertically suspended loads into vertically supported loads, and we have all the data necessary to determine the true arch curve. It will be seen, however, that the curve will depend both upon the magnitude and the distribution of the load, and it follows that for the same span and rise there will be a number of different curves, corresponding to different loads. This question of the load upon an arch has always been the indeterminate quantity which has complicated the solution of the problem, since the volume of material carried by any arch is governed by the line of the curve, and hence the true load cannot be given beforehand.

The error introduced by this indeterminate element is eliminated by the method given below. As previously noted, the horizontal tension in the catenary is constant for all points in the curve, and hence it follows that in the inverted catenary the horizontal thrust must also, in like manner, be constant, and if determined for any one point it will be the same for all other points.

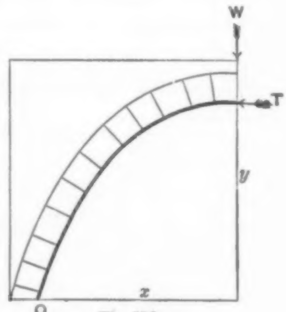


Fig. 159

Now, in Fig. 159 let us take one-half of an arch and examine the forces acting at the crown to maintain equilibrium.

Taking the point of springing O as the centre of moments, we have the vertical force W acting downward at the crown, tending to cause the arch to rotate about O, with a lever-arm equal to x making its moment $= Wx$. Now, if we are to have equilibrium, we must have a horizontal thrust, T, at the crown which has an equal moment. The lever-arm of the thrust is the ordinate y, and hence to have equilibrium we must have

$$Wx = Ty \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1.)$$

For the crown we have $x =$ the half span of the arch and y = the rise, so that if W be given we may at once determine the horizontal thrust T, which, as before stated, is to be the constant for all points in the curve. As already shown, the actual

weight cannot be determined at this stage of the problem, for the limiting curve which governs the exact volume to be supported is not yet known, and it is this point which has rendered previous discussions of the arch of such little practical value. Instead of attempting to determine the exact value of W, however, we may proceed as follows (Fig. 160):

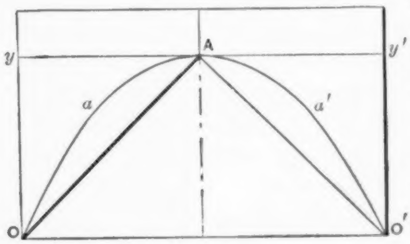


Fig. 160

For the given span and rise of the proposed arch, draw the two inclined lines AO AO' from the crown to the springings; also draw the horizontal line limiting the upper edge of the load. Now assume the load to extend down to the inclined lines AO AO', thus giving two triangles, yOA $y'O'A$, and the rectangular area above as the total load on the proposed arch, and calculate the value of T, the horizontal thrust at the crown under these conditions. We will then get a value of T which will be greater than the true value for the completed arch, because of the excess of load in the segmental spaces between the lines AO AO' and the curves AaO $Aa'O'$.

Solving equation (1) with respect to y we get

$$y = \frac{Wx}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2),$$

and the value of T just found will appear in the denominator of the fraction.

Now, again, let us measure our vertical loads at successive points down to the same triangular outlines for values of W, remembering that in the catenary the vertical component is equal to the weight at any point, together with the sum of all the weights below that point, and that conversely the vertical component on an arch at any point is equal to the load at that point, together with the sum of all the weights beyond that point.

The values of W thus determined will also be in excess of the true values by the excess of area in the same segmental spaces, but now the excess in W appears in the numerator of the frac-

tion (2), and so offsets the excess in the value of T, and hence the correctness of the values of y will not be appreciably affected.*

The method may best be shown by example. Suppose an equilibrium curve be required for an arch of 5 feet span and $2\frac{1}{2}$ feet height, with a level load line 6 inches above the crown, as in Fig. 161.

The first point is to determine the weight W at the crown, in order to deduce T, the constant horizontal thrust. Proceeding as already described to measure the weights down to the line AV, we find that the load is in two figures: the rectangle 6 5 FV and the triangle A 5 V. The area of the rectangle is equal to

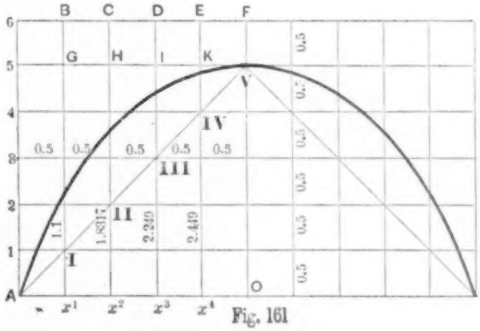


Fig. 161

1.25 square feet, one-half of which acts downward at 5 and one-half at V. The area of the triangle A 5 V is equal to 3.125 square feet, and as its centre of gravity is one-third distant from A and two-thirds from V, two-thirds of the weight goes to A and one-third to V. This gives a total area acting at V of one-half the rectangle 6 5 FV and one-third the triangle A 5 V, or

$$\frac{1.25}{2} + \frac{3.125}{3} = 1.666$$

superficial feet; and this multiplied by the thickness of the arch and by the weight of a cubic foot of the material would give the weight at V in pounds. As this would be proportional to the area, we may use the latter in our calculations.

* As a matter of fact, the error occasioned in the value of y by measuring to the triangular outlines does not appear in most cases until the second decimal place.

Substituting this value of W in the equation

$$T = \frac{Wx}{y},$$

we get

$$T = \frac{1.666 \times 2.5}{2.5} = 1.666 \text{ units.}$$

In this particular case the vertical load and the horizontal thrust are the same, but that is only the case when the height of the arch and the half span are the same. This value of T will remain constant for all points in the curve, and hence we have only to determine the value of W at any other part of the arch, and we can deduce the ordinate at that point.

If we divide AO into five equal parts we may erect ordinates and calculate weight areas correspondingly.

For the fourth ordinate we have:

Overhanging trapezoid $E\ IV\ V\ F$	= 0.375
One-half rectangle 4 6 $E\ IV$	= 1.000
One-third triangle A 4 IV	= 0.666
	2.041 = W_4.

For the third ordinate:

Overhanging trapezoid $D\ III\ V\ F$	= 1.000
One-half rectangle 3 6 $D\ III$	= 1.125
One-third triangle A 3 III	= 0.375
	2.500 = W_3.

For the second ordinate:

Overhanging trapezoid $C\ II\ V\ F$	= 1.875
One-half rectangle 2 6 $C\ II$	= 1.000
One-third triangle A 2 II	= 0.166
	3.041 = W_2.

For the first ordinate:

Overhanging trapezoid $B\ IV F$	= 3.000
One-half triangle 1 6 $B I$	= 0.625
One-third triangle A 1 I	= 0.041
	3.666 = W_1.

The values of x, or the lever-arms of these weights, are :

$$Ax_4 = 2.0$$
$$Ax_3 = 1.5$$
$$Ax_2 = 1.0$$
$$Ax_1 = 0.5$$

Substituting these values in (2) we get :

$$y_4 = \frac{W_4 x_4}{T} = \frac{2.041 \times 2}{1.666} = 2.449 \text{ ft.} = 29.38 \text{ ins.}$$

$$y_3 = \frac{W_3 x_3}{T} = \frac{2.5 \times 1.5}{1.666} = 2.249 \text{ ft.} = 26.98 \text{ ins.}$$

$$y_2 = \frac{W_2 x_2}{T} = \frac{3.041 \times 1}{1.666} = 1.8317 \text{ ft.} = 21.98 \text{ ins.}$$

$$y_1 = \frac{W_1 x_1}{T} = \frac{3.666 \times 0.5}{1.666} = 1.10 \text{ ft.} = 13.2 \text{ ins.}$$

Since the excess of weight due to the inclusion of the segmental space between the curve and the line AV appeared both

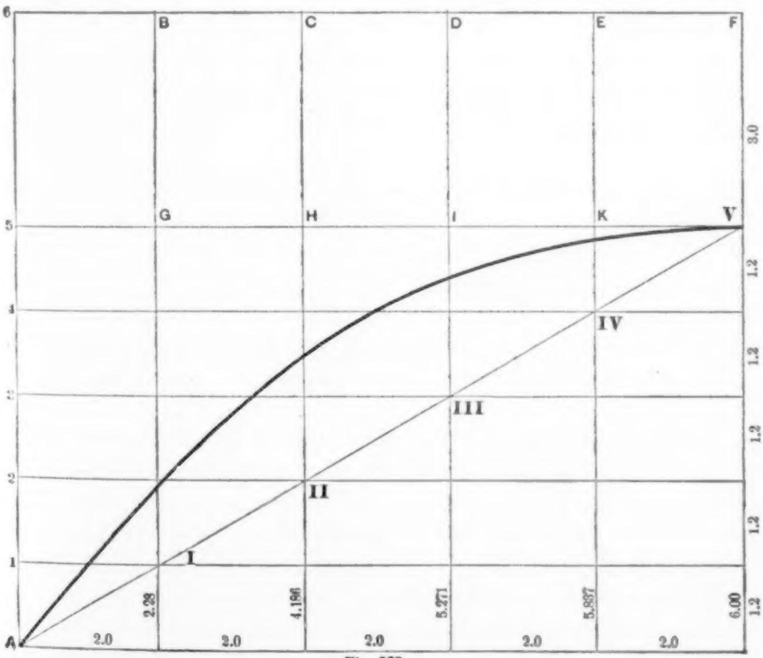


Fig 162

in the thrust and in the loads, it was practically eliminated from the result, and the ordinates are those of a catenary loaded in the same manner as the completed arch. The ordinates calculated by Rankine's method for the same conditions are: 2.44 feet, 2.23 feet, 1.82 feet, and 1.12 feet, which shows the closeness of the approximation.

Another example for a flatter arch will show the general character of the method.

Suppose an arch of 20 feet span and 6 feet high, with a level load line 3 feet above the crown of the curve.

In Fig. 162 we draw the base line and centre line and join the diagonal AV, and determine the horizontal thrust as follows:

The weight at the crown is:

$$\begin{aligned} \text{One-half rectangle } 5\,6\,F\,V &= 15 \\ \text{One-third triangle } A\,5\,V &= 10 \\ &\overline{25} = W. \end{aligned}$$

$$T = \frac{Wx}{y} = \frac{25 \times 10}{6} = 41.66.$$

Now, erecting 4 ordinates spaced 2 feet apart, and drawing horizontal lines through the intersections with AV, in order to assist in measuring the areas, we have:

$$y_4 = \frac{8\,[(E\,IV\,V\,F) + \frac{1}{2}(4\,6\,E\,IV) + \frac{1}{3}(A\,4\,IV)]}{41.66}$$

$$= \frac{8 \times 30.4}{41.66} = 5.837 \text{ ft.}$$

$$y_3 = \frac{6\,[(D\,III\,V\,F) + \frac{1}{2}(3\,6\,D\,III) + \frac{1}{3}(A\,3\,III)]}{41.66}$$

$$= \frac{6 \times 36.6}{41.66} = 5.271 \text{ ft.}$$

$$y_2 = \frac{4\,[(C\,II\,V\,F) + \frac{1}{2}(2\,6\,C\,II) + \frac{1}{3}(A\,2\,II)]}{41.66}$$

$$= \frac{4 \times 43.6}{41.66} = 4.186 \text{ ft.}$$

$$y_1 = \frac{2\left[(B\,I\,V\,F) + \frac{1}{2}(16\,B\,I) + \frac{1}{3}(A\,1\,I)\right]}{41.66}$$

$$= \frac{2 \times 47.566}{41.66} = 2.28 \text{ ft.}$$

The corresponding values by Rankine's method are 5.82 feet, 5.28 feet, 4.17 feet, and 2.46 feet, showing that the approximation is quite close enough for all practical uses.

The use of a few models confirms the deductions above given in a very interesting manner. By hanging blocks of wood cut to the shape of the triangular outlines to a light chain, the catenary may be observed as shown in Figs. 163 and 164, and the

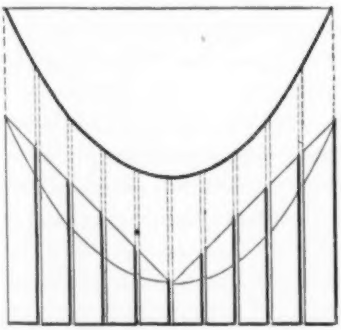


Fig. 163.

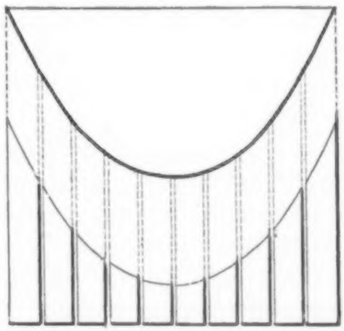


Fig. 164.

removal of pieces cut to the figure of the equilibrium curve produces no appreciable change in the figure of the curve. Model arches built to semi-circular lines and to equilibrium lines for the same conditions show most clearly the superior stability of the equilibrium curve, and an arch built of very shallow voussoirs, if constructed to the proper curve, will stand quite firmly without any cementing material or any "backing" whatever.

In arches built upon equilibrium curves, with the joints at all points normal to the curve, no bending moment or tendency to open at the joints can occur, and the material is subject only to compression as in a vertical wall. The theory of the "middle third," which is sometimes opposed to the use of equilibrium curves, does not properly apply at all under the above conditions, as it was deduced entirely from conditions which only obtain when the joints are not normal to the equilibrium curve.

Instead of drawing an arch to a curve selected arbitrarily and then attempting to investigate the position of the curve which its load may produce, the curve should first be drawn for the height, span, and load given, and then any desired form which will include the true curve within its figure may be used with safety. For sewers and for the construction of subways the equilibrium curve is by far the strongest and most economical, and objections against its appearance to the eye do not hold.

NOTE.—The above method is due to Mr. Thomas J. Lovegrove, of Philadelphia, and is included in patents granted to him for a method of arch construction, in the United States and in Europe.

DISCUSSION.

Prof. Olin H. Landreth.—On the ninth page of the paper is a single point I would like to ask information on. Near the bottom of the page is the following sentence:

"The theory of the 'middle third,' which is sometimes opposed to the use of equilibrium curves, does not properly apply at all under the above conditions, as it was deduced entirely from conditions which only obtain when the joints are not normal to the equilibrium curve."

The "middle third," as I understand its use in arch pressures, has reference not only to a provision for a defect of the arch curve, but provides for the fact that the load is not constant but movable, and therefore the equilibrium curve, or curve of resistance, is not itself constant. If the arch is built for a certain equilibrium curve, the equilibrium curve and arch curve will coincide so long as the load is constant, but if we bring on a concentrated load the equilibrium curve is altered, while the arch curve is not materially altered. Hence it follows that the "middle third" of the arch ring should be wide enough in extent to contain the altered equilibrium curve for all positions of the load. I do not see that this is affected by the method used in finding the position of the equilibrium curve, nor by the direction of the radial joints so long as these do not differ from the normals to the equilibrium curve by angles greater than the "*angle of friction*" for that particular material.

Mr. Supler.—I should like to make one response to Prof. Landreth's remarks. The middle-third theory, as I understand it, is intended to apply to arches where the thrust is at an angle not a right angle with the face of each voussoir, and that

requires that the voussoir shall be of sufficient depth to prevent the line of resultant from passing out of the arch ring.

That theory is of course true wherever the arch is not an equilibrium curve for the entire load, and of course the joints cannot be in the proper position for more than one curve, and in applying the method given to an arch with a movable load, it is proposed to establish first the curvature for a stationary load, then place the desired moving load at various points and determine the deviation in the curve, and make the ring of sufficient depth to include the distorted curve, but it is not necessarily the middle third of the ring. It is an amount which will vary by the extent of the movement or its ratio to the dead load. Therefore it is not a third, or fourth, or tenth, or any definite proportion.

Prof. Landreth.—I would like to put on the board a few lines which would make clear the fact that the equilibrium curve should be contained within a definite one-third of the depth of arch ring, and that the *middle* third. I assume that the curve of equilibrium has been computed and found to be where shown by the dotted line *A B* in Fig. 225. Assuming that the arch has been built symmetrically in regard to that line, the arch curve and the equilibrium curve coincide, and the pressure on all points of each voussoir joint will be uniform and may be represented by ordinates from the line *C D* to the line *H I*, which is approximately a straight line and parallel to *C D*. The total pressure on the joint *C D* may likewise be represented by the area of the rectangular figure *C D I H*. If, however, from any cause, the arch curve has not been properly designed, or if the load has been altered, the equilibrium curve will lie above or below the arch curve, according to the position of the load and the form of the arch. We will assume that the equilibrium curve drops down to the limit between the first third and second third of the voussoir joint, say at *F*; then the pressure on the different points of the voussoir joint will be represented by the ordinates to the line *J K*, and the total pressure on the joint will be represented by the area of the triangular figure *J L K*. In other words, the pressure on the portion of the joint at *J* will be zero, and the

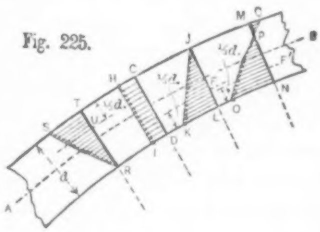


Fig. 225.

pressure on the portion at *L* will be double that of the uniform pressure *D I*, since the area of the two figures must be the same. Hence, by removing the equilibrium curve from the centre of the joint to the bottom of the middle third, we double the pressure on the lower end of the joint and reduce it to zero on the upper end. Suppose you drop the curve still farther to *F*, you produce a diagram of this kind: *M, N, O, P, Q*. You have a pair of triangles now, with their common vertex at *P*, and the pressure on the joint at *M* is now changed to tension, a stress wholly unsuited to masonry. Hence the limit below which the equilibrium curve ought not to fall is where the pressure changes to tension. It is quite safe to say that the arch does not fall at that point, but the critical point is this, that the position of the equilibrium curve not only ought not to be permitted to reach the point at which the arch will fall, but should not pass the point at which the pressure on either end of any joint becomes a tension. And the same thing will similarly follow if the equilibrium curve is raised to the upper end of the middle third at *U*. It will then produce a stress triangle, *R S T*, showing double the uniform pressure at the upper end of the joint *R* and zero pressure at the lower end *T*. You have a lane or a "kernel," as the Germans call it, within which the curve must be maintained in order to have no tension at any part of the joint. So that the value of one-third is a *definite* fraction.

Prof. Webb.—I understand Prof. Landreth to say that the arch would not fall until the line passed outside of the lower limit of the arch.

Prof. Landreth.—I was assuming that the material was sufficiently strong to withstand crushing up to the point of overturning.

Prof. Webb.—Is it true that any material could be sufficiently strong for that?

Prof. Landreth.—No; but I made that assumption since I was discussing stability against rotation and tension, and not stability against crushing.

Prof. Webb.—But for any real material, would it really fall as soon as it got out of the middle third?

Prof. Landreth.—No; it would not, unless the material had a very low crushing strength.

Prof. Webb.—If it would not fall why should the crack open at the top?

Prof. Landreth.—Because, on account of the compressibility of the material, the voussoirs would be compressed in direct proportion to the pressure sustained at different points of the joint, and as this compression is zero at the point P, the two voussoirs while being compressed undergo a slight angular rotation about the point P, which opens the joint at the top (see Fig. 226). An open joint does not necessarily indicate instability, but it indicates a bad distribution of stresses over the joint surface, and, moreover, offers an opportunity for the admission of moisture.

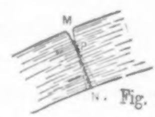


Fig. 226.

Mr. Suplee's models * exhibited at the Cincinnati meeting present a striking ocular proof of the superior stability of arches built on equilibrium curves over circular arches—a fact admitted by all engineers. The opening portions of his paper would give one the impression that the chief reason why arches are not more generally built on equilibrium curves lies in the difficulty in determining the form of the theoretical equilibrium curve. Were this impression correct, it would indicate a low degree of theoretical ability throughout the profession, a conclusion few will admit.

It is true there are many causes incident to construction which operate to make it difficult to cause the *actual* line of resistance to occupy the position intended for it, or even to determine the *actual* position of the line after construction; *e.g.*, voussoir joints of unequal thickness, mortar of unequal consistency, pebbles in the mortar of a voussoir joint, or unequal or lateral settling of either abutment, all render it impossible to locate a given line of resistance *with precision*, but with ordinary care these deviations are usually unimportant, and are neither less nor greater in equilibrium arches than in circular arches.

There are, however, two or three reasons, independent of the above, why circular arches are preferred to arches built on strictly equilibrium lines. These are: 1st. Economy and convenience in construction of both the centring and the masonry arch ring make it desirable that all points of the arch, or of a considerable portion of it, shall have the same curvature in order to permit interchangeability of parts. It is on this account that circular oval arches of three or five centres are preferred to elliptic arches. 2d. The æsthetic requirements of grace and

* Added after adjournment.

beauty are more fully realized in arches of constant curvature, or in arches whose curvature grows flatter toward the crown than in the equilibrium arch whose curvature grows sharper toward the crown. 3d. In arches built to support any material like earth, gravel, or sand, which has the property of exerting some horizontal pressure, the horizontal components of the load acting on the haunches of the arch make the equilibrium curve but *slightly different* from a circle, and hence the adoption of a circle in conformity with the two first reasons above causes a lesser departure from the equilibrium curve than in Mr. Suplee's models, which were designed to sustain and sustained only loads which were strictly vertical. In this connection it is proper to say that the method proposed gives equilibrium curves for vertical pressure *only*, and neglects the horizontal components of the pressure of the load, an assumed condition which is very rarely if ever realized in sewers, tunnels, culverts, or ordinary arch bridges, in nearly all of which material is to be supported whose horizontal thrust amounts to from one-fourth to three-fourths of its vertical pressure.

Mr. Suplee.—I would like the indulgence of the floor for just a few moments to correct a misapprehension and also to correct my own previous statement. Referring to Professor Landreth's statement that I claimed that the error due to the segmental areas was entirely eliminated, I think he will notice in the paper that I said it is approximate only.

Referring to a question asked as to the use of this uniformly loaded curve, it does possess a practical use in determining the curve for a braced arch. Of course you have to take into account the wind stress which would have to be assumed, which would make an unequal loading. For an arch such as that of the roof of the Paris Exposition Machinery Hall, with a joint at the crown and a joint at the spring, it was noticed there that where there was a sudden curve on the roof they were obliged to rivet on many plates. But if that roof had been constructed on an equilibrium curve the thickness of the rib would have been governed solely by the necessity of resisting the thrusts and also such lateral wind pressures as they may have assumed.

I really do not at all differ with Professor Landreth in regard to the variation of the position of the curve in the moving load, and state here that the theory of the middle third does not apply under the above conditions, namely, those of a standing

vertical load. That refers, of course, to arches in buildings. I am quite willing to concede that he is right about the model. The semicircular arch is not an equilibrium curve. If, however, the equilibrium curve is first drawn and the semicircle is made within and without, and the joints made normal to the equilibrium curve, the stability will be greatly increased, and it will require much greater distortion before the pressure becomes disproportionate at any one point in the voussoir.

I should like also to speak of some photographs which I have which show the fact that a pendant load does draw a chain into the same curve as the formula gives for the arch, and that the load measured to the diagonal lines, as shown in the paper, practically gives the same curve.

The arch shown in the photograph is made of dry brick, five feet span, without any cementing material, and is standing perfectly solid.

CCCXCV.

COMPARATIVE TEST OF A HOT-WATER AND A STEAM-HEATING PLANT.

BY R. C. CARPENTER, LANSING, MICH.

(Member of the Society.)

THE test which is described in the present article was conducted by L. R. Taft, Horticulturist of the Michigan Experiment Station and Professor of Horticulture at the State Agricultural College. The test was undertaken to decide whether the better method of

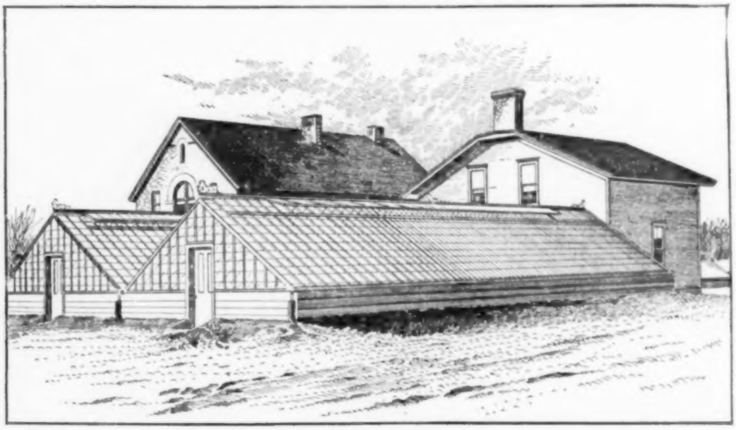


FIG. 165.

heating green-houses was with low-pressure steam or with hot water. The experiment, although undertaken with this especial object, was so well considered and so carefully carried out as to have an important bearing on the general merits of these two methods of heating. At my request Professor Taft consented to allow the presentation of the results to the American Society of Mechanical Engineers, and has afforded valuable assistance in the preparation of the paper. The writer of the article acted as consulting engi-

neer, as regards the construction of the apparatus and the methods of testing.

FIG. 166.

The requirements, to make a fair trial of these two methods of heating, must evidently be, first, two buildings exactly alike; second, two heaters, having the same amount of heating surface and grate

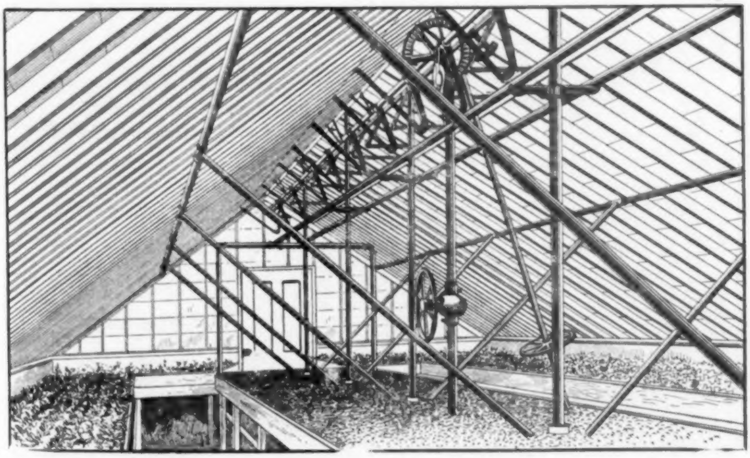
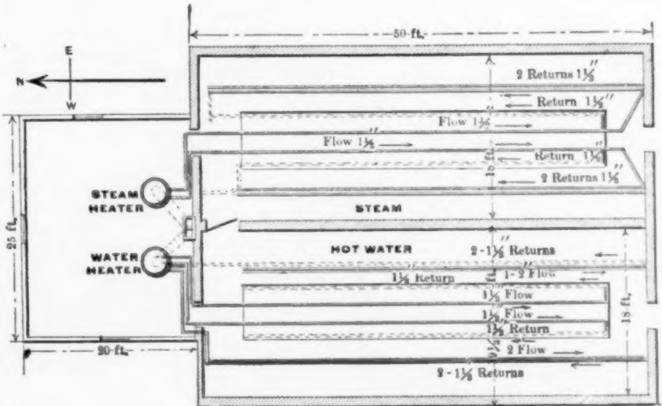


FIG. 167.

surface, arranged as nearly as practicable in the same way; third, a well-arranged system of piping, having the proper surface for the

system under consideration; fourth, every essential feature of a perfect system of heating of each class.



GROUND PLAN

FIG. 168.

The buildings (Fig. 165) were erected for this especial test, each green-house being 50 by 19½ feet, placed side by side, extending north and south. The hot-water plant was used to heat the most westerly house, which in our locality is the most exposed to cold

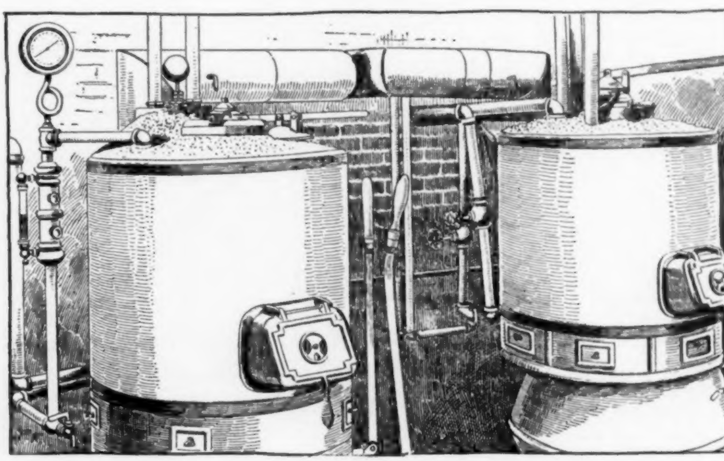


FIG. 169.

winds, and consequently, under ordinary circumstances, the most difficult to heat. The past winter, however, there has been little

difference in this respect. Each house contains eleven hundred square feet of glass; the side walls were built of concrete from the foundation to the level of the ground outside; above this wooden walls of double thickness, with air space, extend two feet high, making the total height of the side walls on the inside of the building four feet. The rafters consist of permanent sash bars, 2½ inches

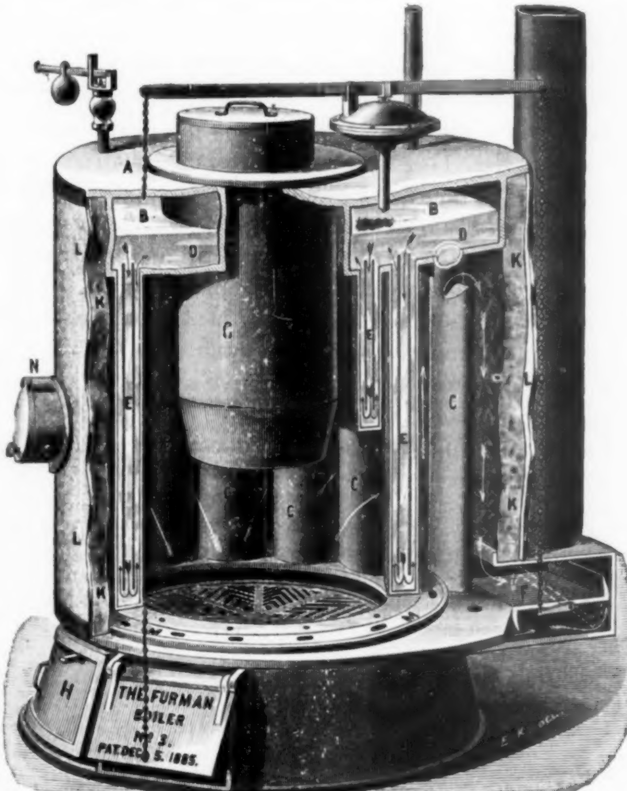


FIG. 170.

deep, and one foot apart, bearing on their upper surface double strength glass. They are supported in place by a truss built of one-inch gas-piping, which shows clearly in the views presented, Figures 166 and 167. Both buildings are ventilated by windows in the roof, hinged to the ridge, and are raised by elbow-joint fixtures made for this especial purpose.

To the north of the houses above described is a two-story addi-

tion, 25 by 20 feet, with a cellar. The heaters for the work were located in this cellar, as shown in the accompanying plan (Fig. 168), and were connected to a chimney containing two flues, each 8 by 12 inches—each heater was connected to a separate flue by an 8-inch circular pipe of sheet iron.

The heaters adopted for this work were each of the Furman No. 2 type, made by the same manufacturer, Herendeen Manufacturing Co., and are nearly identical, each having the same amount of heating surface, grates exactly the same size, and external dimensions the same.

The heaters are shown by Fig. 169, as they were fitted for the test. The cut, Fig. 170, shows the internal construction of the steam heater. The hot-water heater is the same, except for a partition in the water chamber directly over the centre of the tube *E*, so that the return water must pass downward and upward in these tubes before it can possibly enter the circulating pipes. It is to be noticed that nearly the entire amount of heating surface is derived from drop tubes, connected with the water-chamber only in the upper part of the boiler. An inside tube or a partition provides for a continuous circulation. (Fig. 172.)

The heating surface in the heater in question was as follows:

	sq. ft.
13 drop tubes, each 26 inches long, 1 foot in circumference......	28.17
1 drop tube, 13 inches long, 1 foot in circumference.............	1.08
3 drop tubes, each 2 feet long, 2 feet in circumference...........	12
Top of heater, 2 feet in diameter effective.........................	3.14
Surface of magazine exposed to flames..........................	1.57
Total.........................	45.96
Deduct area magazine reckoned effective.........................	.78
	45.18

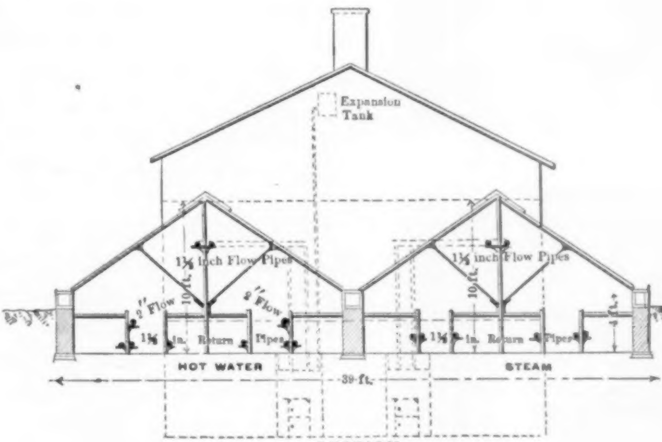
The area of grate surface is 1.89 square feet, or in the proportion of 1 square foot of grate surface to 23.8 square feet of heating surface.

The System of Piping.—This is shown in the accompanying sketch (Fig. 171), and is arranged as follows:

1st. For the steam-house. Two main pipes, 1½ inch diameter, rise to the height of two feet below the ridge, extend with slight fall to the south end of the green-house, each pipe then supplies three return pipes 1½ inch diameter, laid with a fall toward the heater. Two of these return pipes are supported by the legs of a

side bench next the walk, the third by a leg of a centre bench; see Fig. 171. These pipes are supplied with automatic air-valves, of the Jenkings type. The radiating surface of the steam pipes is almost exactly 200 square feet, the cubic contents of the building is 6,174 cubic feet, giving the ratio of 1 foot of radiating surface to $30\frac{7}{8}$ cubic feet of space, or 1 to $5\frac{1}{2}$ square feet of glass surface.

2d. For the house heated by hot water. In this case, there are four main supply or flow pipes, two of them are $1\frac{1}{2}$ inch diameter, and two are 2 inches in diameter. The smaller flow pipes are



SECTION

FIG. 171.

arranged exactly like the main pipes in the steam-house, but each supplies a return pipe of the same size, carried on the legs of the middle bench. The 2-inch flow pipes rise a few feet above the heater, and then extend to the further end of the house, being supported by the legs of the side benches; each of these supplies two $1\frac{1}{2}$-inch return pipes, placed underneath the flow pipes. In all cases the flow pipes rise to full height directly on leaving the boiler, and then descend uniformly throughout the remaining distance. The radiating surface of the pipes in the hot-water house is 275 square feet, giving the ratio of 1 foot of radiating surface to 25.4 cubic feet of space, or 1 to 4 square feet of glass surface.

The pressure on the hot-water system is maintained uniform by an open expansion tank, 12 inches diameter by 18 inches high

located in the second story of the addition, 18 feet above the water in the lowest returns.

The steam system is supplied with pressure gauge safety-valve, and a diaphragm check draught and damper regulator. The hot-water system is supplied with an automatic damper regulator, designed by the writer of this paper. The apparatus in all cases worked satisfactorily.

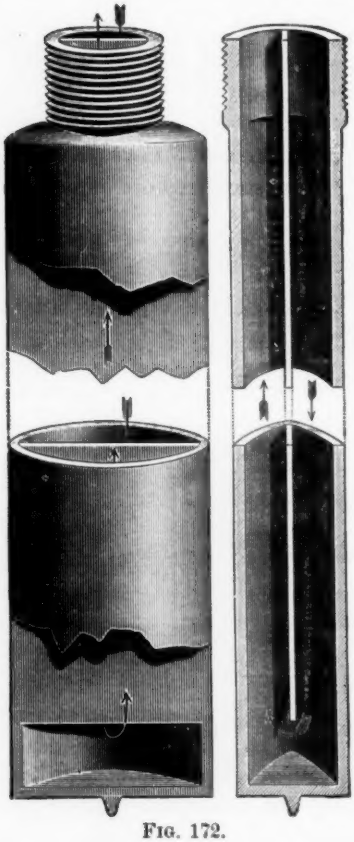


FIG. 172.

The steam system was put in operation about November 15, 1889, and used steadily since that time. This system was considered very perfect and economical by the attendant, Mr. Miller, an experienced green-house man having it in charge.

The hot-water system was put in operation about December 10th, and has been in use steadily since.

The test of the two heaters was started December 21st and continued to March 1st. The method of testing was as follows: the heaters were fired regularly at 6 A.M., at 4 P.M., and at 9 P.M., the coal being added in lots of 25 pounds. As both heaters were furnished with magazines, this method was in every instance practicable. It does not, however, make it possible to compare the exact consumption of coal day by day in the different heaters, as the amount remaining in the magazine at any one time was not measured. When the time, however, becomes considerable, this error becomes exceedingly small, and soon practically disappears. The test shows the amount added day by day to each heater—which is very nearly the daily consumption, as the magazine capacity was only about 100 pounds each.

Temperature observations were taken in each house and outside, at 6 A.M. and 9 P.M. Maximum and minimum readings for the day

were also observed. The thermometers used for this purpose were carefully standardized before commencing the test, and in each case were suspended in the air, so as not to be influenced by surrounding objects.

Those in the green-houses were suspended in the same relative position in each house, on one side, about one and a half feet from the roof, and separated from any heating pipes by a bed of growing plants. The results show the decided economy of the hot-water heater, as compared with the steam heater. A surprising result, to the writer at least, was that this difference (not the per cent.) increased rather than diminished with extreme cold weather.

The variation or extreme range between maximum and minimum seems to have been about the same with either heater, although the opinion of the florist was decided that the interval between fires could be much longer with the hot-water heater than with the steam heater. Before commencing the test, the florist in charge considered steam heat more economical and better adapted for green-houses than hot-water plants. The following table gives the temperature observations, and the chart following shows the outside temperature, temperature in each house, and coal consumption for each day. The temperature maintained was all that was required by the vegetation growing in the respective houses.

TEMPERATURE OBSERVATIONS AND COAL CONSUMPTION, DECEMBER, 1889.

Date.	Outside Temperature.			Hot-Water House.			Steam-Heated House.				Coal Consumption, lbs. per Day.	
December, 1889.	6 A.M.	4 P.M.	9 P.M.	6 A.M.	9 P.M.	Maximum.	6 A.M.	9 P.M.	Maximum.	Minimum.	Hot-Water Heater.	Steam Heater.
21.......	28	44	43	56	58	58	45	50	50	42	100	125
22.......	41	43	32	55	56	60	54	54	61	51	75	75
23.......	34	40	40	54	58	60	52	54	54	52	75	75
24.......	38	62	58	63	65	65	56	68	68	52	75	75
25.......	38	39	41	57	52	70	53	58	67	53	25	50
26.......	34	32	33	51	50	57	56	52	62	....	100	100
27.......	24	40	22	50	55	....	50	52	54	48	75	100
28.......	24	36	38	52	54	60	50	52	57	....	50	75
29.......	34	33	26	64	52	64	58	48	63	51	75	100
30.......	20	24	12	51	51	53	55	53	56	47	100	150
31.......	14	30	32	51	56	56	53	53	57	50	75	100
									Total.	....	825	1,025
Average.	31.8	38.5	35.1	54.9	55.2	60.3	53.9	54.0	....	....	75	93.2

During the eleven days of testing in December, the hot-water heater consumed, on the average, 75 pounds of coal per day, the

steam heater, 93.2 pounds, or 24.3% more coal. It is also noticed that the temperature was maintained higher in the hot-water house. Both heaters were used as surface-burners. No draught regulator attached to hot-water heater.

TEMPERATURE OBSERVATIONS AND COAL CONSUMED, JANUARY, 1890.

Date.	Outside Temperature.			Hot-Water House.					Steam-Heated House.					Coal Consump., lbs. per Day.	
January, 1890.	6 A.M.	4 P.M.	9 P.M.	6 A.M.	9 P.M.	Maximum.	Minimum.	Extreme Range.	6 A.M.	9 P.M.	Maximum.	Minimum.	Extreme Range.	Hot-Water House.	Steam House.
1	38	51	51	58	48	60	......	......	56	48	58	51	7	50	50
2	46	36	33	57	50	64	......	......	51	55	60	50	10	75	75
3	20	30	30	47	48	53	......	......	42	47	57	40	8	75	75
4	30	38	40	55	60	60	......	......	42	60	60	42	18	75	75
5	44	56	56	60	62	63	53	10	62	62	64	54	10	50	50
6	46	32	26	62	53	66	......	......	59	52	63	59	4	75	100
7	24	28	20	47	54	56	46	10	52	50	52	47	5	100	125
8	24	28	22	49	52	55	46	9	52	52	54	47	7	125	125
9	20	36	36	51	51	55	51	4	50	52	54	47	10	75	100
10	36	38	36	54	55	55	53	2	57	53	60	52	8	50	75
11	36	49	40	53	54	56	49	7	51	53	60	51	9	50	75
12	34	42	62	50	52	56	48	8	52	55	65	47	8	50	100
13	28	26	20	49	52	57	45	12	49	53	63	47	16	125	125
15	......	30	32	49	55	55	45	10	50	53	57	47	10	75	100
14	34	34	32	55	54	57	50	7	53	54	57	52	5	75	100
16	16	16	12	53	57	62	49	13	50	54	55	48	7	125	150
17	12	28	26	56	55	58	52	6	54	53	55	50	5	100	125
18	26	30	24	56	56	56	53	3	53	54	54	52	2	100	125
19	26	40	40	55	54	58	53	5	58	53	58	52	6	75	100
20	30	24	20	54	56	56	50	6	52	[illegible]	54	51	3	125	150
21	16	6	-3	54	55	57	53	4	53	55	55	53	2	175	225
22	-3	16	14	55	55	55	53	2	52	55	55	52	3	150	175
23	16	26	22	55	55	55	54	1	53	56	56	52	4	100	150
24	12	19	14	54	55	56	53	3	53	54	57	53	4	125	175
25	26	34	35	53	56	56	52	4	52	57	56	52	4	100	125
26	36	44	42	55	57	58	53	5	54	56	57	50	7	50	50
27	32	38	28	55	55	58	53	5	51	55	57	50	7	100	125
28	20	35	28	55	57	57	53	4	52	56	58	52	6	100	125
29	34	40	36	57	58	58	54	4	53	53	58	53	5	75	100
30	24	36	40	54	57	57	52	5	52	55	55	51	4	100	125
31	48	36	30	60	60	60	57	3	58	54	58	54	4	75	100
Per day Average.	27.7	......	27.2	54.1	54.[illegible]	......	......	5.8	52.5	53.8	....	....	6.5	90.3	112.1
Average after January 17th.								4.0	......				4.4	......	

The heaters were used as surface burners principally until January 17th. Beginning with that date the magazines were kept nearly full. The draught regulator on the hot-water heater was first put in operation at this time. The range or variation in temperature after that date, was, for the hot-water heater, 4 degrees F., and for the steam heater, 4.4 degrees.

The average coal consumption for the month was 90.3 pounds per day for the hot-water heater, and 112.1 pounds per day for the steam heater, the steam heater using for the month 24.003% more

coal. It is also to be noticed, that the temperature of the hot-water house is maintained somewhat higher.

February	Atmosphere			Hot-Water House.					Steam House.					Coal.	
	6 A.M.	4 P.M.	9 P.M.	6 A.M.	9 P.M.	Maximum.	Minimum.	Range.	6 A.M.	9 P.M.	Maximum.	Minimum.	Range.	W. H.	S. H.
1..	24	30	22	54	55	58	52	6	50	54	54	50	4	100	125
2..	28	34	38	58	56	58	55	3	55	56	57	54	3	50	100
3..	40	41	38	60	56	60	56	4	58	54	58	56	2	100	125
4..	40	62	60	60	65	60	55	5	57	65	57	53	4	50	75
5..	32	28	18	57	56	65	54	11	54	54	65	53	12	125	150
6..	18	24	22	56	55	58	54	4	55	55	57	53	4	100	125
7..	24	28	24	56	56	57	54	3	55	56	60	54	6	100	125
8..	14	20	16	54	56	56	54	2	52	56	56	52	4	125	175
9..	10	24	26	54	55	57	52	4	54	55	57	54	3	125	175
10..	22	20	28	55	57	57	53	4	55	56	57	54	3	100	125
11..	20	43	32	55	56	57	55	2	56	57	56	56	0	100	125
12..	24	32	31	53	54	56	52	5	52	56	57	52	5	100	125
13..	22	42	32	55	54	57	53	3	52	57	56	52	4	75	100
14..	38	44	36	55	55	57	53	4	56	55	58	56	2	75	100
15..	22	36	24	51	56	56	47	9	55	54	57	55	2	100	150
16..	30	48	32	53	57	56	......	......	53	55	55	52	3	75	100
17..	36	50	40	60	57	60	55	5	56	57	61	55	6	75	75
18..	33	32	24	54	56	62	52	10	52	56	57	52	5	125	125
19..	18	32	26	56	58	57	54	3	54	56	56	53	3	100	100
20..	14	17	10	54	54	58	50	8	56	55	57	50	7	150	175
21..	8	19	14	50	56	56	50	6	52	55	58	50	8	150	175
22..	18	30	28	55	56	57	54	3	53	54	60	53	7	100	100
23..	32	34	24	54	55	56	53	3	50	55	56	50	6	75	75
24..	34	47	34	54	56	57	53	4	56	56	56	55	1	75	100
25..	32	36	30	57	54	57	54	3	57	54	57	56	1	125	125
26..	28	28	30	58	56	58	54	4	53	56	57	51	6	75	75
27..	26	39	32	56	58	56	56	1	55	56	56	53	3	75	100
28..	30	26	16	54	56	58	53	5	50	56	57	50	7	150	175
Av.	22	....	....	54.4	56	57¼	53.2	4½	54.1	53.5	57.6	53	4⅓	99.1	121.4

Per cent. of coal used by steam heater over that by hot water, 22.5.

The weather during February was quite mild, not falling at any time below 8 degrees. The relative consumption of coal remained much the same as in the previous months, the hot-water heater using on the average 99.1 pounds of coal per day, the steam heater using 121.4 pounds per day.

So little extreme cold weather has been experienced during this test, that the question is still open as to the relative economy of the different methods of heating under such circumstances. So far as can be premised from the observations recorded, the per cent. of gain of hot water over steam would be somewhat less under such circumstances.

So far as variation or range of temperature is concerned, the two heaters were practically the same. This is perhaps a test of the automatic damper regulators more than of the heaters, but in no case did this range exceed 4 degrees, in either heater, when the fire was carefully attended to. A greater range in every case denoted

the fact that the coal supplied to the furnace at that particular time had been too small.

RECORD OF TEST OF HOUSES, MARCH 1ST TO 20TH, 1890.

	Temperature.											Coal Burned.	
	Atmosphere.			Hot-Water House.				Steam House.					
March.	6 A.M.	4 P.M.	9 P.M.	6 A.M.	9 P.M.	Maximum.	Minimum.	6 A.M.	9 P.M.	Maximum.	Minimum.	Hot Water House.	Steam House.
1	8	15	14	56	54	56	55	55	54	56	53	125	150
2	10	20	18	58	58	58	58	56	......	56	52	100	125
3	18	30	26	58	64	60	58	55	60	56	53	200	200
4	22	16	8	53	55	65	53	53	57	61	53	150	150
5	3	10	3	53	50	56	53	50	54	56	50	150	200
6	8	18	10	53	56	54	50	53	58	54	52	150	200
7	3	26	16	45	54	58	55	57	57	58	53	125	125
8	6	29	19	48	58	54	47	47	57	57	47	125	150
9	12	40	30	57	58	59	51	49	55	59	49	125	150
10	34	36	36	56	60	63	55	55	61	60	53	75	100
11	38	52	48	58	60	60	58	55	58	61	55	50	50
12	32	39	34	55	60	60	54	54	58	58	52	75	100
13	30	40	34	50	60	60	50	52	53	60	50	100	100
14	34	27	16	56	54	60	56	54	52	55	53	125	125
15	12	34	12	54	55	55	53	50	52	55	50	150	175
16	12	28	26	58	58	58	55	56	56	57	55	125	175
17	28	38	32	53	58	58	53	54	56	60	53	75	100
18	28	36	30	54	58	58	52	54	54	56	52	75	100
19	30	56	28	53	56	56	50	56	54	56	54	75	100
20	30	......	......	56	......	56	55	52	......	57	52	......	......
											Average:	114.4	135.7

NOTE.—The low temperature of the houses March 7th, 8th, and 9th, was occasioned by neglect of fires occasioned through sickness of the regular attendant.

During the extreme cold weather of March 5th, 6th, 7th, and 8th, the steam heater consumed 168¾ lbs. of coal per day, the hot-water heater 137.5 lbs. per day. For this period the steam heater consumed 22.7% more coal than the hot-water heater, but for the whole month only 18.6% more coal.

APPENDIX.

During the month of April, 1890, both heaters were supplied with the same amount of coal daily, with the result as shown in the following table, from which it is seen that, with the same consumption of coal in each heater, the temperature of the hot-water house was 8.5° higher than that of the steam house, as shown by averaging the maximum and minimum temperature for obtaining the mean temperature in each case.

TEST OF HOT-WATER HEATING PLANT AND STEAM-HEATING PLANT, APRIL, 1890.

April, 1890.	Temperatures.															Pounds of Coal.	
	Atmosphere.			Hot-Water House.					Steam House.							
	6 A.M.	4 P.M.	9 P.M.	6 A.M.	9 P.M.	Maximum.	Minimum.	Range.	6 A.M.	9 P.M.	Maximum.	Minimum.	Range.	Water House.	Steam House.
1	16	36	20	60	60	63	60	3	52	54	58	50	8	100	100
2	20	46	40	60	65	63	58	5	52	56	54	50	4	100	100
3	44	43	38	65	65	66	65	1	55	58	56	55	1	75	75
4	46	38	32	64	50	65	62	3	60	56	60	58	2	100	100
5	26	52	40	58	60	58	57	1	55	55	56	53	3	75	75
6	42	56	52	64	60	65	60	5	55	57	58	55	3	75	75
7	44	56	48	64	60	64	58	6	55	56	57	53	4	75	75
8	48	72	58	62	65	62	60	2	54	60	56	50	6	25	25
9	38	40	38	58	60	68	58	10	50	56	60	48	2	75	75
10	26	38	36	60	60	60	59	1	52	53	58	52	6	100	100
11	30	65	54	57	64	60	55	5	53	60	55	50	5	25	25
12	56	74	64	62	67	64	60	4	56	65	63	56	7	50	50
13	54	68	52	65	68	67	65	2	58	61	65	56	9	25	25
14	38	54	40	60	62	68	60	8	50	54	66	48	8	75	75
15	32	47	34	58	60	62	58	4	50	50	54	50	4	75	75
16	26	60	34	58	60	60	58	2	50	58	52	48	4	50	50
17	30	60	40	56	60	60	56	4	51	52	60	51	9	50	50
18	32	38	30	57	56	60	55	5	48	50	56	48	8	75	75
19	26	48	38	57	56	57	56	1	50	50	54	50	4	50	50
20	28	64	38	55	58	56	54	2	50	54	54	50	4	50	50
21	30	68	48	55	60	60	55	5	47	55	55	47	8	25	25
22	40	70	58	55	65	60	55	5	46	62	56	46	10	50	50
23	56	58	60	63	65	65	63	2	59	60	62	59	3	50	50
24	34	56	34	55	60	64	54	10	46	48	60	44	14	50	50
25	30	56	42	54	58	60	54	6	46	50	48	46	2	50	50
26	36	42	40	55	55	60	54	6	45	50	54	45	9	75	75
27	36	60	46	55	52	58	54	4	53	50	53	47	6	50	50
28	30	60	54	54	60	56	51	5	46	58	50	44	6	25	25
29	52	56	42	54	56	60	54	6	55	50	58	55	3	50	50
30	40	64	46	52	60	58	52	6	47	48	55	46	9	50	50
Average..		36.20	......	58.40	60.23	61.63	57.33	4.30	51.86	54.86	54.93	49.03	5.90	60	60

Average of Maximum and Minimum.... 59.48........................... 51.98

Fig. 173.

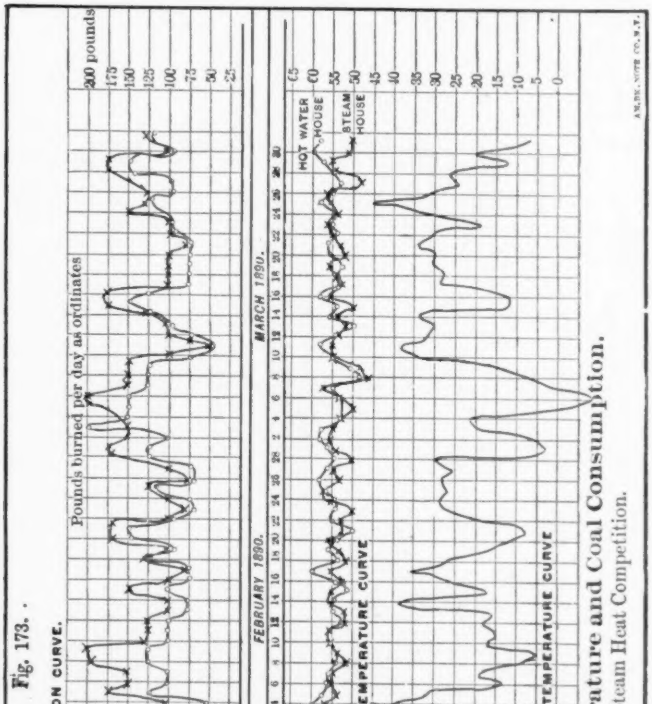


Chart showing Temperature and Coal Consumption.
Hot Water and Steam Heat Competition.

DISCUSSION.

Mr. Charles E. Emery.—It is believed that hot water is in general much better adapted for heating green-houses than steam, on account of the greater specific heat of water and the greater weight of heated fluid in circulation which tend to produce equality of temperature under average conditions of management. It must not, however, be assumed, either from general principles or the results of experiments like those recorded in the paper, that there should be any difference in the *economy* of the two systems. The difference in the quantities of heat leaving and returning to the boiler in order to maintain like green-houses at the same temperature under the same conditions, when using two different fluids, must of necessity be the same, and such a difference as is shown by these experiments must be entirely due to the difference in the efficiency of the heaters themselves, when used under different conditions. Tracing out in detail the differences in condition, we find, first, that the amount of radiating surface in the building heated with steam was considerably less than in that heated with water. This made no difference in the number of thermal units utilized, but required that the average *temperature* of the steam used in one case should be greater than that of the water in the other. Taking it for granted that the two heaters were exactly alike in all essential particulars, the water-heating surface of the one used for steam was necessarily slightly reduced by the necessity of lowering the level of the water to provide steam space. The water and steam were also necessarily at a somewhat higher temperature in the steam heater than in the water heater. These two elements would necessarily increase the amount of coal burned in any steam heater compared with that in any water heater when the conditions varied as stated. Moreover, in heaters of the general type illustrated it is impracticable to control the draft areas so as to force the heated products of combustion over the entire heating surface, from which fact it frequently occurs that a very slight increase in demand from such a boiler causes a great increase in the amount of heat carried away to the chimney. The steam pressure and actual temperature of the hot water circulated are not given, but could not vary the conclusion that experiments of this kind are valuable only as showing the relative economy of the particular heaters employed, operated under particular con-

ditions, and that from such results conclusions cannot be drawn as to the performance which would be obtained with other apparatus for the same purpose.

Mr. Carleton W. Nason.—While Mr. Carpenter's interesting paper is important as working in a field of experiment in which there has been but little comparative research, it is to be regretted that he has omitted in his data several figures which would have largely increased the value of the record, not only as guiding us to conclusions as to why the results reached were obtained, but as showing, under the conditions named, that the economy in favor of the hot-water system might have been predicated.

The most important omissions are as follows:

1st. Average temperature of the water in circulation.

2d. Average pressure of steam used.

3d. Temperature of gases in the chimney-throat of both boilers.

4th. Average temperature of the gases in the two fire-boxes before coming in contact with the fire surface of the boiler.

Assuming, however, that the following figures, which are about what are in practice commonly used, the cause of the economy becomes at once evident.

Temperature of the water surface, 160 degrees.

Temperature of the steam surface, 216 degrees.

What the temperature of the fire-boxes is we do not know; but taking the largest average comsumption of coal for 24 hours, which is given for the steam-boiler for the month of March, 1890, as 135.7 lbs., we have the remarkably low consumption of 6.07 lbs. per hour, or 1.89 square feet—the grate area—into this weight, of 3.21 lbs. per square foot per hour. Now, with so slow a fire I think I am not much out of the way in assuming that the smoke temperature will not greatly exceed 400°.

Looked at from one standpoint, the fire surface of a boiler may be considered as a sort of " strainer " placed there for the purpose of extracting heat from the gases on their way to the chimney, and the greater the difference in temperature between the gases and the cooling surface, the finer, so to speak, the strainer becomes, or, in other words, the cooler the gases are in entering the chimney—leaving more heat behind them.

In this instance, assuming that the figures I have taken are

approximately correct, we would have gases at 400° coming in contact with surface at 216° in the steam-case, and in that of the hot water at 160°, or a difference between the two of 56°; and as the fire surfaces in the two boilers are equal in area, it is, therefore, evident that in the chimney-throat of the hot-water boiler a lower temperature of about 56° should naturally be looked for than would be found in that of the steam.

Assuming again that the temperature of the air passing under the grate bars is 60°, this would give 400° — 60° = 340°, as the degrees of temperature actually raised in the fire-box. Now, the 56° difference in temperature between the steam and hot water surface is 16½% of this amount in favor of the hot water, or very nearly what is claimed by Mr. Carpenter.

That this line of reasoning is correct is proved by assuming that the rate of combustion be reduced to as low a point as to develop a fire-box temperature of not more than 210°; then the results from the steam-boiler would be *nil*, while the hot-water boiler would still be to a certain extent effective, delivering its water at a temperature of about 50° lower, or, say, at 110°.

The more rapid the combustion or the higher the gas temperature, the smaller the percentage of difference between the temperature of the fire surface becomes when compared with it, and the less difference it makes whether steam or hot water is used.

In the case cited, both boilers are abnormally small for the work to be performed, and consequently the discrepancy between them is greater than would be commonly looked for.

In green-house practice, where the fire is carelessly looked after, or the external temperature fluctuates rapidly, there are certain advantages obtained by the use of steam, from the way in which the heating surfaces can be rapidly heated or cooled—there being stored in the apparatus only the specific heat of the iron surfaces, instead of those of a large body of hot water in circulation in addition, in which case green-house temperatures must be partially controlled by opening ventilators, and heat from hot water thus lost. But with boilers of slow combustion these results are evidently more than offset by the low temperature of fire surfaces in hot-water boilers.

If Mr. Carpenter has the figures I have alluded to as not given it would be interesting at a future meeting to have them, in order to learn how nearly they correspond with those I have assumed and to have their results substantiated.

Mr. J. L. Gobeille.—There is no doubt in the minds of all who have been in a position to see different systems of heating in practice that hot water is far more efficient, because of a far more equable temperature, and is more economical than either steam or hot air. At the Nashville meeting you may remember that this subject came up for quite a good deal of discussion. It was my expression then that in a very few years we would see, especially in the northern parts of the country, a decided change in regard to hot water. That change has come about. I will venture to say that in Canada one-half of the domestic heating is done with hot water; and I will venture to say further that in Canada, during the past year, there have been taken out more steam and hot-air systems from dwelling-houses, and water heating substituted, than there were water-heating plants in operation at the date of that meeting. I believe that this experiment of Professor Carpenter's is one of very great value; and that it comes about as near being correct in practice as anything which is done in that way—notwithstanding the apology which he makes in the beginning. Almost every condition has been fulfilled to give us an idea of the practical working of the two systems side by side under almost the same conditions, and I doubt whether we will see another experiment so valuable to persons putting in heating plants as this one is. It is certainly something of great value to the society; and I wish at the next meeting Mr. Carpenter might give us further results, just as he has these, so that we might study them and compare them with our own practice.

Mr. D. W. Robb.—As a native of Canada, I am able to endorse what has been said by the last speaker. I agree in the main with the discussion which has taken place on this paper—that, theoretically considered, steam heating should give as economical results as hot-water heating, but in practice it does not hold good. In Canada I have had something to do with both steam and hot-water heating. I am entirely disinterested and have no object in using one system more than the other, and have been compelled in a number of instances, from purely practical considerations, to use hot-water heating.

Mr. Carpenter.—I desire just to say one or two things. I would say, in the first place, that I have very recently found another report of a similar trial made by the Massachusetts Experimental Station at Amherst, and from it I find that they have ob-

tained even a greater difference than I have. They had very different boilers, but had their two green-houses arranged somewhat similar to ours. I am not certain in regard to the details of that test. We took observations of the temperature of the escaping gases, but did not take them regularly, and not being fully satisfied with the results of those observations, did not give them. The greatest temperature shown by these measurements of the escaping gases was, from the steam heater about 280°, and from the hot-water heater about 200°; the greater proportion of these measurements, however, indicated an average temperature about 40° less than the results given above. The temperature of the out-flowing water was taken a number of times, and usually did not exceed 160°, although in one instance we found it as high as 190°. The temperature of the return water was from 15° to 45° below that of the out going water. These observations I did not consider very valuable, from the fact that we did not take them as systematically as should have been done, and as I should have liked. I had expected to make an evaporative test of the steam-boiler, but could not find time. The system of hot-water heating of green-houses in which 4-inch cast iron pipes were used is hardly as economical as steam heating. We have a green-house which has been heated with hot water put in on that system for years. I may say that that green-house, where the pipes are 4-inch cast iron, took more than double the amount of fuel this last winter per foot of area heated than this green-house put upon the modern system. It is my experience that until the new system was put on the market the green-house men were throwing out the old-fashioned way of heating green-houses by hot water, and were putting in steam. Now the tide has set the other way, and hot water is going back again.

I* desire to say in regard to the questions asked by Mr. Emery, that an examination of the structure of the two heaters will show that there is no sensible difference between the heating surface in the steam heater and in the hot water heater. The reason for this statement will clearly appear on examination of the drawings submitted. The pressure carried on the steam heater averaged rather less than two pounds per square inch, the safety-valve being set to blow off at five pounds. The only pressure on the hot-water system was that due to the position of

*Author's closure, under the Rules.

the expansion tank, as explained in the article. It might have been, for the main coils, six pounds of pressure. The temperature of the hot water seldom reached 160°, and so far as our observations could tell, never equalled 190°. I fully agree with Mr. Emery as to the reason for the relative economy of the hot water as compared with steam, and I think that this difference will generally be found in ordinary systems of heating, since the hot water is used at a much lower temperature than the steam.

CCCXCVI.

NOTES ON KEROSENE IN STEAM-BOILERS.

BY R. C. CARPENTER, LANSING, MICH.
(Member of the Society.)

THE paper on this subject by L. F. Lyne, published in Vol. IX., page 247, induced me to give the oil a trial in the boilers of the Steam Heating Plant of our State Agricultural College at Lansing, Mich.

The boilers experimented on are of the ordinary tubular type, each being 12 feet in height, four are 4 feet in diameter each, and the remaining two each 5 feet in diameter.

The use of the oil was commenced in April, 1889, with the boilers very badly incrusted with a hard scale. Previous to this time the only method used to remove the scale had been by employing a workman to go inside the boiler and knock off what could be reached with a hammer and scaling iron. The estimated cost of this method of cleaning was $18 to $25 per annum for each boiler, and the results were very unsatisfactory, as more than two-thirds of the heating surface was entirely inaccessible. The scale was fully three-eighths of an inch thick in many places.

The first application of the oil was made while the boilers were being but little used, in the warm season, by inserting a gallon of oil, filling with water, heating to the boiling point and allowing the water to stand in the boiler two or three weeks before removal. By this method fully one-half the scale was removed during the warm season and before the boilers were needed for heavy firing. We found, however, that a more effectual way was to apply the oil in small quantities when the boiler was in actual use. There is no question regarding the action of kerosene oil in softening the particular scale deposited from the water which we use (an analysis of that water is appended), and it also seems to work in between the scale and the boiler plate in such a manner as to loosen large flakes of the scale, some-

times fully a foot in length. On opening the boiler they are found deposited on the bottom, and readily removed by rakes; the great mass of the scale, however, is removed as a fine mud, through the blow-off pipe.

In regard to the quantity to be used, we found that beyond a certain amount no benefit seemed to result; for boilers 4 feet in diameter and 12 feet long we obtained the best results by the use of 2 quarts, or one-half gallon, for each boiler per week, and for each boiler 5 feet in diameter 3 quarts per week.

The oil was readily fed by adding a one-fourth inch branch to the suction pipe of the feed-pump, and leading it to a vessel containing kerosene oil. By this methed a quantity as large or small as desired could be introduced into the boiler at the same time with the usual feed, and in the form of an emulsion of water and oil.

At the present writing our boilers have less scale in them than at any previous time during the past four years, and the small amount running in them seems to be soft and gradually disappearing. A large portion of each boiler shows the clean black steel, apparently in as good condition as when new.

I do not believe that the oil will produce any injurious effect on the iron—in fact, I cannot see why it is not the best possible preservative for it.

Our boilers are at some distance from the various buildings heated, and steam is transported through underground pipes; the extreme distance being about 800 feet in opposite directions. Despite the small quantity of kerosene used in the boilers, the odor was perceptible by opening an air valve to any steam radiator in any of the buildings.

When as much as a gallon per week per boiler was used, the odor was very strong, but with one-half that amount it was hardly perceptible, and only to be noticed when an air valve had been open a long time.

Since commencing to use the oil, a much greater deposit of rust scales than usual has been found in the various steam-traps in the buildings, indicating, in my opinion, that the oil is also exerting a cleansing influence on the pipes of our whole system.

The expense of this oil is very light, as compared with any other preparation for cleansing boilers known to the writer. The expense to the college per boiler has not exceeded $2 per annum for the kerosene used.

The water used in our boilers is obtained from a flowing well having a depth of 260 feet. The following analysis, by Dr. R. C. Kedzie, shows its composition:

$Ca\ CO_3$, (Carbonate Calcium)	206	parts in 1,000,000.
$Mg\ CO_3$, (Carbonate Magnesium)	78	" " "
$F_2\ CO_3$ (Carbonate Iron)	22	" " "

Traces of sulphates and chlorides of potash and soda.
Total solid parts, 325 to 1,000,000.

NOTE.—The oil used was refined kerosene and not crude petroleum.

DISCUSSION.

Mr. Henry A. Porterfield.—It may be of interest to some who have been troubled with scaling boilers to state that some of the purges which are used and sold under the name of resolvents and all sorts of names are crude petroleum, practically, at a very advanced price. I have no doubt that many of you who have used them know that. I would like to state that at the next meeting I hope to be able to make a report on the use of a mechanical boiler cleaner. This boiler cleaner is to be put on a brand new boiler in a place where the scale is very difficult to keep down. Kerosene and petroleum and all sorts of things have been tried, and nothing has proved perfectly successful. The use of one grade of petroleum seems to do very well, but it does not do away with the difficulty entirely. Kerosene does not alone. The appliance I refer to is called the Hornish mechanical boiler cleaner. If any gentlemen present have any experience with it or any other mechanical appliance, I should be very glad to hear from them.

Mr. Carleton W. Nason.—I had a personal experience very similar to that spoken of by Mr. Carpenter. Some two or three years ago I bought a boiler, 5 feet by 16 feet, horizontal tubular, second hand. It had been used, I think, for about a year in Brooklyn at a brewery. The boiler was in perfectly good condition, except that it was scaled on the inside to a thickness from one-half to three-quarters of an inch. I put in crude petroleum for about a month, using about one-half the quantity which Mr. Carpenter says was used in the boiler with which he is familiar. The result was entirely satisfactory. The boiler was cleaned in from four to six weeks, and the scale, instead of coming off in small fragments, came off in large flakes. The boiler is now

as good as new. The oil used was such crude petroleum as is sold by Charles Pratt & Co. I do not know what grade it was, or where it came from. The result was quite satisfactory.

Mr. A. F. Nagle.—Why did Mr. Nason use the crude petroleum instead of the kerosene, as recommended by Mr. Lyne as being the better in his experience?

Mr. Nason.—It was convenient to take the advice of the foreman of the shop, who said that some one had told him it was a good thing to use.

CCCXCVII.

LENGTH OF AN INDICATOR CARD.

BY J. BURKITT WEBB, HOBOKEN, N. J.
(Member of the Society.)

In a paper * at the last meeting I called attention to the fact that the motion of an indicator drum is governed by the same law which controls the motion of a simple mass attached to a spiral spring. It was also shown that the drum and drum-spring have their natural time of vibration, which could, if desirable, be made variable at will, either by altering the length of the spring or using interchangeable springs of different strengths, or by varying the moment of inertia of the drum by attaching other masses to it, or otherwise.†

The amplitude of the drum motion was also touched upon and it was shown by a simple arithmetical argument that if the engine forces the drum to oscillate in less than its natural time, the amplitude of its oscillation will be increased by an amount dependent on the elasticity of the cord or other connection; and, in the same way, if the time be made greater the amplitude will be decreased.

This paper supposes the previous paper to have been read, and continues the subject by taking up more at length the question of the change of amplitude.

To make the treatment of the problem more general, the supposition that the spring b is a very stiff spring will be abandoned (see Fig. 174, which is the same as Fig. 47 of Paper CCCLXXIII.) and, to indicate that no such distinction is made between them, the springs will hereafter be called S_1 and S_2.

Neglecting the masses of S_1 and S_2, and also the slight bend-

* Paper No. CCCLXXIII.

† Some indicators are so constructed that the drum-spring may be tightened a will; this has, however, no effect upon the time of vibration.

ing* of the wire, two spiral springs can differ in their mechanical action only by exerting different forces for the same amount of extension.

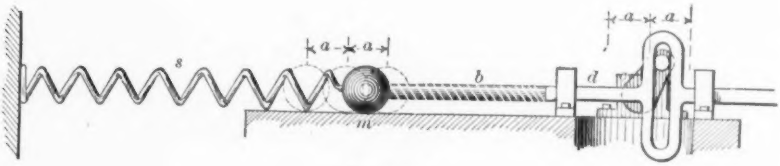


FIG. 174.

OSCILLATION BY A SINGLE SPRING.

Before discussing the oscillation of a mass by two springs the action of a single spring should be had clearly in mind. This is explained in works upon mechanics with more or less care and is treated in Rankine's *Applied Mechanics*, p. 494, under the head "542. Straight Oscillation."

Let O (Fig. 175) represent the face-plate of a lathe upon which a mass m is guided to slide in a radial direction without friction. Let the spring S_1 be attached to the face-plate at D so that its natural length brings the centre of the mass to the centre, O, of the face-plate, as shown dotted.

When the lathe is not running m will remain at rest at O and to maintain it in any other position A, a distant from O, the lathe must be run at such a speed as will produce an exact balance between the centrifugal force, F', of m and the tension, T_1, of S_1.

If ω is the angular velocity† of the face-plate the centrifugal force of the mass is

$$F' = ma\omega^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and, if f_1 is the number of pounds required to stretch the spring one foot (or ten times that required to stretch it one-tenth of a foot), the tension for the distance a is

$$T_1 = af_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

* In a spiral spring subjected to tension or compression, although the main tendency is to twist the wire itself, there is a slight tendency to bend it, which increases with the pitch and is nothing for zero pitch.

† The angular velocity is the linear velocity of a point at one foot from the centre.

Equating these three results for equilibrium, when the centrifugal force due to the mass of the spring is neglected,

$$F' = T_1 = ma\omega^2 = af_1 \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or

$$\omega^2 = \frac{f_1}{m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

As T_1 is proportional to a, it may be represented by the arrow marked T_1.

From the fact that a cancels out of the equation it is clear that, if the angular velocity of the face-plate be made equal to

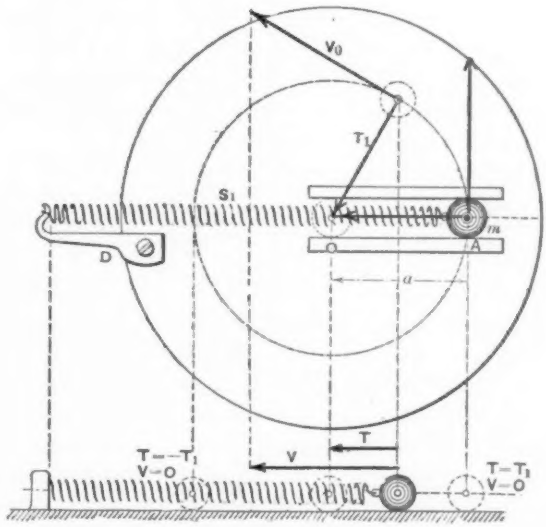


FIG. 175.

$\sqrt{f_1} \div \sqrt{m}$, the mass will remain balanced at any distance from the centre, a, at which it may be placed. For the time of one revolution, under these circumstances, there results from the definition of angular velocity:

$$t_0 = \frac{2\pi}{\omega} = 2\pi\sqrt{\frac{m}{f_1}} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

When a mass is oscillated in a right line by a spring, neglecting the mass of the latter and other disturbances, the result is a simple *harmonic oscillation*, which is neither more nor less than

the horizontal projection of the motion of the revolving mass, and this projection furnishes all the facts of the oscillation.

The following are the correspondences in detail:

The *semi-amplitude*, a, of the oscillation equals the radius a.

The time of a complete oscillation is t_0, the time of a revolution, and is the same for any amplitude.

The motion of the oscillating mass is such that it is always directly beneath the revolving one.

The extreme tension (or compression) of the oscillating spring equals T_1, and the tension at any time is T, the corresponding horizontal projection of T_1, and this is, therefore, the accelerating force which controls the motion of the mass. It is easily seen from this that this force is, as it should be, proportional to the distance of the oscillating mass from its mean position, or, what is the same thing, to the elongation of the spring.

The velocity V at any time is the horizontal projection of the velocity V_0 of the revolving mass, therefore,

The greatest and least velocities are V_0 and 0.

OSCILLATION BY TWO SPRINGS.

Suppose that a second spring, S_2, be added, as shown in Fig. 176. Let it be of such a length that the mass, when attached to S_2 only, will have its centre at B, distant b from O, as shown by the dotted circle. Suppose also, as before, that when m is attached to S_1 only its centre will be at O. When both springs are attached the mass will occupy an intermediate position, C, distant c from O, the value of c depending upon the strengths of the springs. This position must be that in which the two springs pull with equal force, and, the elongations of S_1 and S_2 being respectively $OC = c$ and $BC = b - c$, this leads to the equation

$$\text{Force} = cf_1 = (b - c)f_2 \quad . \quad . \quad . \quad . \quad . \quad (6)$$

which reduces to

$$c = \frac{f_2}{f_1 + f_2} b \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

The position C is, therefore, that in which the mass will be at rest when the lathe is not running. If now the lathe be put in motion centrifugal force will drive the mass further from O,

and if the angular velocity be made equal to the value previously found for it, viz., $\sqrt{f_1} \div \sqrt{m}$, the mass will evidently take the position B, at which the tension of S_2 is zero. For greater velocities the mass will be beyond B and for less it will be between C and B.

Let a be the distance of the mass from the centre for any angular velocity ω, then the equilibrium between the tensions and the centrifugal force is expressed by the equation

$$af_1 = (b - a)f_2 + ma\omega^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

which reduces to

$$a = \frac{f_2}{f_2 + f_1 - m\omega^2} b \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

This equation shows what has been stated already, that when $m\omega^2 = f_1$, that is, when the centrifugal force is balanced by S_1 alone, $a = b$, while for less and greater values of ω a will be respectively less and greater than b. For $a > b$ S_2 will be compressed while for $a < b$ it will be stretched.

The mass is represented at the same distance from the center as in the preceding figure, but that it may be in equilibrium at this point the velocity must be enough less (as indicated by the shorter arrow) to allow for the pull exerted by S_2.

To understand the motion of the mass in Fig. 174 it suffices to consider the horizontal projection of the arrangement just described, as was done in the previous case.

In the horizontal projections, Fig. 176, a harmonic oscillation is supposed to be given to the piece B' so that its end B' oscillates with an amplitude $2b$ about the central position. This piece takes the place of d, Fig. 174, its amplitude being called $2b$ in place of $2a$. The first and last horizontal projections show the extreme positions, while the third shows the middle position, where the springs have their natural lengths; the second projection shows an intermediate position. The velocity and accelerating force at any point can be found as before by horizontally projecting the force and velocity of the revolving mass. The arrow representing the force can still be drawn from A to O by using a different scale to that previously employed, and it will represent the difference, T'', of the tensions of the springs, while the varying length of its projection will represent the changing difference of tensions for the harmonic oscillation.

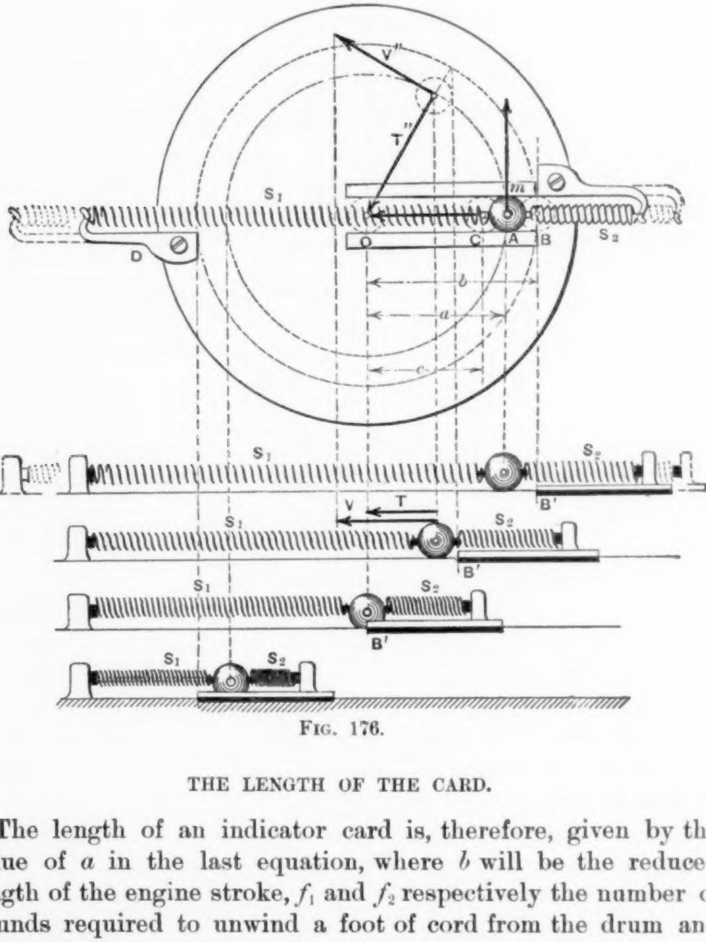


FIG. 176.

THE LENGTH OF THE CARD.

The length of an indicator card is, therefore, given by the value of a in the last equation, where b will be the reduced length of the engine stroke, f_1 and f_2 respectively the number of pounds required to unwind a foot of cord from the drum and to stretch the cord connection a foot (or ten times the force required for a tenth of a foot), ω the angular velocity of the engine, equal to $2\pi \div$ time of one revolution, and m the moment of inertia of the indicator drum divided by the square of the radius of the drum pulley expressed in feet and measured to the centre of the cord.

The only feature yet unexplained in Fig. 174 is that the springs therein are supposed to sustain an extra tension of thirty-two pounds, while in the horizontal projections of Fig. 176, the springs are compressed when m is to the left of the centre. To put such a tension in the device, Fig. 176, nothing more is

necessary than to suppose the supports D and B to be extended, as indicated in dotted lines, by amounts inversely proportional to the strengths of the springs. Thus, let the left-hand end of S_1 be so extended a distance f_2 and the right-hand end of S_2 a distance f_1, then the tension in each spring will be increased by the amount $f_1 f_2$ and, no matter where the horizontally oscillating mass may be, it will not feel the change, because the difference of the tensions will remain unchanged.

By thus stretching the springs at their ends not attached to the mass, and by amounts proportional, instead of equal, to f_1 and f_2, any desired constant tension may be added to the varying tension and compression, so that the springs may be kept in tension throughout the whole oscillation, as is supposed to be the case in Fig. 174.

VALUES USED IN THE FIGURES.

In constructing Figs. 175 and 176 the following are the values assumed:

$$\begin{aligned} f_1 &= 1\,;\ f_2 = 2\,; \\ c &= \tfrac{2}{3}\,;\ a = .8\,;\ b = 1\,; \\ T_1 &= .8\,;\ T'' = .4\,; \\ m &= \tfrac{1}{8}\,; \\ V_0 &= 1.6\sqrt{2}\,;\ V'' = 1.6\,; \\ \omega &= 2\sqrt{2} \text{ and } 2\,; \\ t &= \pi\sqrt{\tfrac{1}{2}} \text{ and } \pi. \end{aligned}$$

In Fig. 176 the dotted extension of S_1 is made twice that of S_2; there was, however, not enough room to indicate sufficiently great extensions to prevent some compression toward the left-hand end of the oscillation.

SECOND METHOD.

Another way of treating the problem, which leads to the same result, will now be explained.

Suppose that the engine revolves in a longer time than the natural time of drum oscillation, so that a is less than b.

Let the spring S_1 be supposed to be divided lengthwise into two springs, S_1' and S_1''. Were S_1 a flat band of rubber a longitudinal cut would so divide it, but for the spiral spring two others of the same length must be substituted. Let their combined strength be the same as that of the original spring, that is, let

$$f_1' + f_1'' = f_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

and let S_1' be made of such strength, f_1', that it alone will cause the mass to oscillate in the same time as the cross-head. Now, suppose S_2 and S_1'' to be disconnected from the mass and attached to each other by a link occupying the same length as the mass, thus leaving m entirely under the control of S_1'; if now the amplitude of the mass be made such that the mass and link remain together—*i.e.*, oscillate with the same amplitude—the problem is solved.

Thus, according to the numerical values already given for Fig. 176, S_1' and S_1'' will be equal springs of half the strength of S_1, for equation (5) becomes

$$t_0 = 2\pi\sqrt{\frac{m}{f_1'}} = 2\pi\sqrt{\frac{1}{8f_1'}} = \pi,$$

or

$$f_1' = \frac{1}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

which, substituted in (10), gives

$$\frac{1}{2} + f_1'' = 1,$$

or

$$f_1'' = \frac{1}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Now, when S_2 and S_1'' are linked together and the other end of the former oscillates with the semi-amplitude b, the link will oscillate with an amplitude dependent on the relative strengths f_1'' and f_2. In fact, if f is the number of pounds to stretch a spring one foot, one pound will stretch it a distance expressed by $1 \div f$, so that, for each pound tension applied at B, the sum of the elongations of the two springs is

$$\textit{Elongation per pound} = \frac{1}{f_1''} + \frac{1}{f_2} \quad . \quad . \quad . \quad (13)$$

But the total elongation for the two linked springs equals the semi-amplitude b, and the semi-amplitude a is the elongation of S_1'', which gives the proportion

$$a : b = \frac{1}{f_1''} : \frac{1}{f_1''} + \frac{1}{f_2};$$

but

$$f_1'' = \frac{1}{4} f_2;$$

therefore,

$$a : b = 4 : 5,$$

or

$$a = .8b = .8. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

When the mass has this semi amplitude a, S_1' and S_1'' will always remain equal in length, so that it is manifestly a matter of indifference whether they be regarded as two separate springs or as united into one.

Where a is greater than b a different supposition is to be made.

Suppose then that S_2 be temporarily removed and another spring S_2', differing from S_2 only in being longer, be attached to the mass and to a fixed point to the right of the sliding piece B'. Let the length of S_2' be such that the combined strength of S_1 and S_2' will be sufficient to oscillate the mass in the same time with the cross-head, *i.e.*, give f_2' such a value as to satisfy equation (5), thus :

$$t_0 = 2\pi \sqrt{\frac{m}{f_1 + f_2'}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where t_0 is the time of revolution of the engine and, therefore, of the oscillation of B'.

For a spring infinitely long f_2' would be zero and it can be increased to any amount by shortening the spring, therefore (15) can always be satisfied.

Now let the fixed point be so chosen that, were the mass connected with S_2' only, it would stand in its central position, just as it would if connected with S_1 only.

Under these circumstances the mass will oscillate in the time desired and, if the proper amplitude a be given to it, the left-hand portion of S_2', between the mass and that point whose semi-amplitude is b, will be identical with S_2, and will be equally compressed. It must be equally compressed because S_2' exerts the same force on the mass as S_2, otherwise the time would not be the same, and some point must have the amplitude b, because $b < a$ and all points of a spring oscillate with amplitudes proportional to their distances from the fixed end.

It is therefore a matter of indifference whether the connection with B' be regarded as made or not, inasmuch as B' and the point on the spring, having the same semi-amplitude b, must remain together anyhow.

To make this supposition correspond as nearly as possible with that made for a less than b, the spring S_2 may be regarded as the difference of two springs S_2' and S_2'', the latter being that part of S_2' which reaches from the fixed point to the point whose semi-amplitude is b; *i.e.*, S_2' may be regarded as the sum of S_2'' and S_2.

DISCUSSION.

Mr. Chas. W. Barnaby.—I have been considerably interested in this paper. From 1883 to 1886 I gave this indicator-drum matter some attention, and in a paper * which I presented at the Chicago meeting in the latter year, under the title of "Another New Steam-engine Indicator," I went over the ground pretty thoroughly, showing how the paper carrier might be placed between two sets of springs, in the way of which the author speaks, and reciprocated at a high speed, when there was necessarily considerable force to overcome, by a cord under very light tension. I will not undertake to go into details at the present time, but will merely refer those interested in the subject to that paper.

Mr. H. H. Suplee.—In regard to the alteration of the length of indicator cards by various disturbing forces, I had an experience in the case of a tandem Corliss engine, where the distance from the reducing motion to the drum of the instrument was quite long —at least 10 feet—and I got such very irregular results, owing to the stretch of the string, that instead of endeavoring to correct for it, I did my best to eliminate it. I succeeded fairly well by running a thin, flexible wire from the reducing motion straight back past the cylinder, and attaching it to a stout spiral spring, thus placing the greater portion of this long connection in the shape of a practically inextensible wire; then I was able to run the cord from the indicator down to an arrangement making quite a short connection. It seems to me that instead of trying to correct these errors, we should try to eliminate them.

Prof. Webb.—The paper is purely a discussion of what the

* Transactions, Vol. VII., p. 489, No. CCX.

effect of the elasticity of the string is, and the limits which I assigned to it forbid a discussion of the advisability and methods of avoiding that stretch. There are a number of ways in which it may be done, and in the paper to be read by Professor Jacobus one is described which was tried successfully with the Pawtucket engine. In any way of avoiding the stretching of the string, care must be taken not to introduce friction, because that produces a change of phase, and invalidates the card to a certain extent. Alteration in length is not a disadvantage if uniform, and the necessity for elimination is only because it may sometimes not be unlform from some disturbing cause.

CCCXCVIII.

SOME EXPERIENCES WITH CRANE CHAINS.

BY C. SEYMOUR DUTTON, YOUNGSTOWN, OHIO.

(Member of the Society.)

At the Washington meeting there was some discussion on the subject of chains used for cranes, in which the opinion was expressed by the writer that the failure of such chains was almost invariably due to the use of bad material in the construction of the chain, and not to any mysterious deterioration of the iron from use. This, of course, has nothing to do with the failure of chains manifestly too light for the use. This opinion was based on a considerable personal experience with wrought iron in construction, beginning with tests of material for the St. Louis Bridge.

During the past year I have had a little experience bearing directly on this subject. We have, in the foundry department of William Tod & Co., a power crane, on which we have occasionally to lift twenty-five tons. The crane was built about fifteen years ago, and equipped with a chain made of 1⅛-inch iron, running over sheaves sixteen inches diameter. The winding drum is twenty inches diameter, the drum and sheaves all being grooved deeply enough to prevent the links touching the bottom of the grooves. The chain had become considerably worn in the bends of the links, and had broken, perhaps twice, but not at the worn places. We ordered a new chain, taking some pains to try to find reliable chain-makers, and to impress on them the necessity for the best material and work. Within two weeks after putting on the new chain it broke under a load not exceeding eighteen tons on the crane, the fracture being short at near the middle of the straight sides of the link. The chain-manufacturers then made another one, but before it arrived the chain in use failed again, under a crane load of not more than twelve tons.

The link first broken was sent away, but in the accompanying

illustration, Fig. 180, is one-half of the link last broken. Fig. 181 is another link from the same chain, cut out and broken to examine the quality of the iron, and Fig. 182 is a link from the chain last made and now in use. This link was cut open hot at one end to remove it from the chain. It was then nicked on the inside of the bend at the other end, and mashed down under a hammer to its present shape. The link first broken presented a fracture similar in appearance to that of Fig. 180.

Now here was a chain which failed twice in the first month of its use, while only two, or possibly three, links had been broken from the old chain in fifteen years' continuous use. The crystallization theory will hardly account for this, although the fractures of the broken links present the same appearance, and are just such as are usually attributed to that popular delusion. The photograph tells the whole story. The iron is cold-short, and totally unfit for the use.

In characterizing the crystallization theory as a delusion I am perfectly aware that iron is often subjected to such severe flexure or vibration, continuously, as to destroy its cohesion and render it weak and brittle; but while a crane chain in constant use is treated rather rudely, it does not receive anything like sufficient punishment to produce such an effect. The fracture would also be quite different in appearance, bright, it is true, but presenting a surface of rounded granules, rather than sharp crystals.

This condition is probably caused by a gradual breaking up of the fibres, without any of the supposed rearrangement as crystals, and is doubtless very weak as well as brittle; while cold, short iron is usually very strong against a direct pull.

A phosphorus determination from borings from Fig. 180 showed .00125, or, as it is usually expressed, .125 of one per cent., which is sufficient to account for the character of the iron.

The last chain in the present case has not failed in any way up to the present time, and if it is all of as good material as the link mashed, Fig. 182, it will probably only give out when worn to an insufficient section at some point.

The question of principal interest to most of us is, how to obtain good chains. I know of but one certain way, and that involves some trouble and expense. I refer to the inspection of the material before manufacture, and of the workmanship after, by a competent engineer.

Giving a piece of chain a mild pull in a testing machine enables the manufacturer to certify that it has been "tested,"

Fig. 180.

Fig. 181.

Fig. 182.

and may occasionally open a particularly bad weld, but can serve no other useful purpose.

The ability to stand bending cold is the most important quality of chain material, because that is the principal, if not the only, cause of chain failure. In fact, for this purpose the

cold bend test will usually show all the engineer wants to know of the material.

Perhaps, as the millennium approaches, a certificate of inspection which shall mean something will accompany all chains. At present, however, we shall have to conduct the inspection at our own expense, or take what we can get. There are, undoubtedly, some manufacturers who would make chain to suit our specifications, if not hampered in price; and there may be some who make reliable work of their own free will; but the search for them is not certain to be short or inexpensive. Our own experience in this direction has not been reassuring. At any rate, we are not likely to get good chain, or good anything else, unless we ask for it. And as long as we are willing to believe that a brittle link was made so by some mysterious influence or condition of its use, so long will chains be made of cheap cold-short iron.

DISCUSSION.

Mr. Gus. C. Henning.—The experience given in the paper under consideration is unfortunately one of too common experience, and the photographs shown and explanations given leave no doubt that the fault of the chain was solely cold-shortness—high phosphorus. This percentage of phosphorus is a very common one in some of the best iron made in Pittsburgh and elsewhere, giving in the tension test as much as from 51–56,000 lbs. tenacity, 25–30 per cent. elongation in 8 inches, and from 35–60 per cent. reduction, accordingly as the material is tested in bars of from $\frac{3}{4}$ inches to $2\frac{5}{8}$ inches diameter, all these factors changing in proportion to the diameter, or work done during rolling. This same iron, however, although welding admirably when using a bright coke or gas fire, will give considerable trouble with coal fire. Furthermore, although this material when it comes from the rolls is thoroughly fibrous, and shows no crystallization whatever, it is precisely of that quality which, when subjected to heating and hammering, or heating and quenching, will become granular or crystalline, and become valueless, especially for chains, every link of which is heated, re-heated, welded, and hammered.

I wish to add my opinion to that of the author, that there is no such thing as gradual crystallization of iron when properly used, although constantly in vibration. The iron shown and described, however, can be injured by shock, or, rather, percussion,

and this can occur in crane chains, which are generally very roughly handled. They often stick under heavy castings or forgings, and then a sledge is used to loosen them; this sledging can in such iron produce granulation and ultimate failure of the chains.

Inasmuch as such abuse is of common occurrence, enough care cannot be exercised in the selection of a low phosphorus and low sulphur iron, nor can the process of manufacture of chains be too carefully watched at all times.

With such iron the workmanship is all important, and even with the greatest care many defects may exist; and inasmuch as smiths are prone to cold-finishing, to use water on hot material, and to straighten material cold by sledging, an iron should be selected which cannot be injured by such treatment. This can be readily done by intelligent inspection, and there would be no difficulty in obtaining a material which will give entire satisfaction, being free from accidental liability to injury by any rough treating short of cutting.

Unfortunately, the makers of chains do not, as a rule, take the proper means to have their material inspected, and generally depend almost entirely upon tension tests and welding tests, and as long as no granulation appears, the material is supposed to be satisfactory. The proper manner to inspect and test this material is to subject it to such treatment as it can be subjected to by rough usage, and then determine whether such handling will injure, or develop dangerous qualities, or injure good ones.

With our present knowledge of materials, processes of manufacture, and accessibility of testing apparatus, there is no excuse for using material which is not of suitable quality for chains, or for any purpose whatever.

The old test of applying a proof-load is of no value whatever, as it does not discover defects which careful inspection could not reveal. Such defects which would be developed by this test only exist when the work has been carelessly, negligently, or cheaply made.

The fracture shown in the paper is distinctly that of a material unfit for chains, although of very excellent qualities for almost any purposes except where welding and hammering becomes necessary, or where accidental injury by percussion is possible or liable to occur.

This material, a fracture of which is shown in Fig. 180, would,

under normal conditions, show a fracture as Fig. 182 when broken as described.

The main trouble in obtaining the proper material would be the price that is asked for it, but as the cost of workmanship is generally greater in chains than that of the material, this should not be of primal importance.

Even considerable abrasion of running chains should not injure the material sufficiently to make it inferior, much less to cause rupture.

Moreover, as electric welding is at the present day a complete success, as far as welding of similar pieces of cylindrical section is concerned, there should be no difficulty in obtaining the most perfect chains. Electric welds are guaranteed to have, under such conditions, fully 95% of the original bar section.

CCCXCIX.

AN AUTOMATIC ABSORPTION DYNAMOMETER.

BY GEORGE I. ALDEN, WORCESTER, MASS.

(Member of the Society.)

THE desirability of maintaining a uniform load upon an engine used for experiments and tests, of accurately measuring the useful power developed by the engine, and of automatically regulating the rate at which energy is absorbed, led to the construction of the device herein described.

This dynamometer is essentially a friction brake, in which the pressure causing the friction is distributed over a comparatively large area, thus giving a low intensity of pressure between the rubbing surfaces. The pressure is produced by the action of water from the city pipes. Enough water is allowed to pass through the machine to carry off the heat due to the energy absorbed. The rubbing surfaces are finished smooth and run in a bath of oil. A valve operated by the slight angular motion of the dynamometer varies the supply of water, and consequently the pressure between the frictional surfaces, thus securing automatic regulation.

Referring to the drawings, *A* (Fig. 185) is an iron disk keyed to the crank shaft *B*. The sides of this disk are finished smooth, and each side has one or more shallow radial grooves as shown at *X* (Fig. 183). The outer shell consists of two pieces of cast-iron, *CC*, bolted together, but held at a fixed distance apart by the iron ring *D*—whose thickness is the same as that of the disk *A*—and by the edges of the copper plates *EE*. Each of these plates at its inner edge makes with the cast-iron shell a water-tight joint by being "spun" out into a cavity in the iron and held by driven rings *FF*. Thus between each copper plate and its cast-iron shell there is a water-tight compartment, *WW*, into which water from the city pipes is admitted at *G*, and passing to the opposite compartment through passages, as shown at *O*, is discharged through a small outlet at *H*.

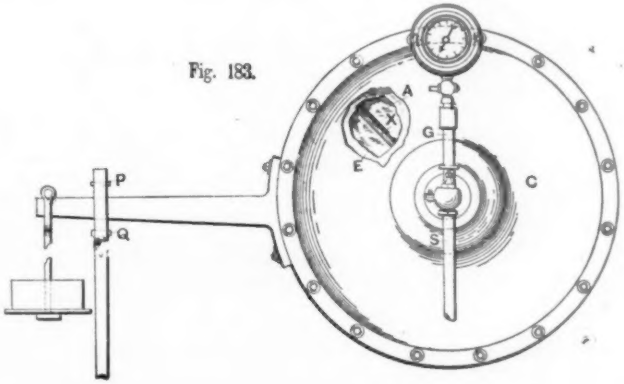
The chamber *MNN* is filled with oil, which finds its way from *N* to *M* along the grooves in the disk *A*.

The shaft is free to revolve in the bearings of the cast-iron shell *CC*. The shell has an arm carrying weights, as shown in Fig. (1). The arm has its angular motion limited by stops at *P* and *Q*.

An automatic valve at *V* (Fig. 184), and shown in sections—Figs. 186 and 187—regulates the supply of water to the machine.

The valve consists of two brass tubes fitted one inside the other, but free to revolve relatively to one another. The inside tube has one end closed. Each tube has slots parallel, or nearly parallel, to its axis. One tube connects with the supply pipe *S*,

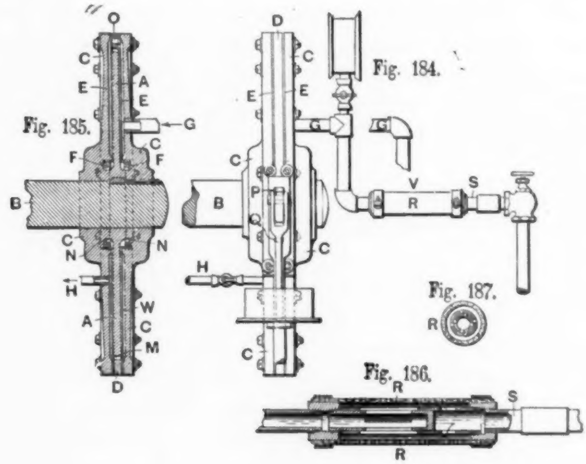
Fig. 183.



the other tube with a pipe rigidly fixed to the brake and communicating with one of the compartments *W*. A flexible tube, *R*, encloses the whole. The valve is so adjusted that a slight angular motion of the brake varies the free water passage through the slots (see Fig. 187); and the aperture at *H*, through which the water is discharged, being small and constant, the pressure of the water in the chambers *W W* is thus automatically varied.

The dynamometer is operated as follows: The chamber *NNM* being filled with oil, weights are suspended from the arm to give the desired load. The engine is started, and when up to speed a valve is suitably opened in the water-pipe leading to the automatic valve (*V*), which latter being open, allows water to pass to the compartments *W W*. The pressure of this water forces the copper plates against the sides of the revolving disk *A*—with which they were already in contact—causing sufficient

friction to balance the weights upon the arm, which then rises. This motion operates the automatic valve, checking the flow of water to the brake and regulating the moment of the friction on the disk to the moment of the weights applied to the arm of the brake. The first trial of the machine gave remarkable results, the arm standing midway between the stops, with only a slight and slow vibration, and this without the use of a dash-pot. The



water seems a little sluggish in its action in response to the motion of the regulating valve, so that there is no sudden vibration of the arm, and the load is practically constant.

The low intensity of pressure at the friction surfaces, their smooth finish and perfect lubrication, a constant temperature maintained by a uniform flow of the cooling water, and the sluggish action of the water in response to the slight motions of the regulating valve, are all favorable to the unusual steadiness and complete regulation realized in the actual operation of the brake.

CCCC

THE KINZUA VIADUCT, 1882.

BY THOMAS C. CLARKE, NEW YORK CITY.
(Member of the Society.)

It has been said that more engineering skill was shown in avoiding the use of large bridges and viaducts on a certain line of railroad, than could possibly have been shown in building them.

Such was not the case with the Kinzua viaduct. A short branch line of the Erie Railroad, called by a name nearly as long as itself—"The New York, Lake Erie and Western Coal and Railroad Co."—runs from the main line at Bradford, Pa., to certain coal fields in McKean Co., Pa., and it became necessary to cross the Kinzua creek, which flows through a deep valley some 300 feet below the level of the surrounding country.

An attempt was made to cross this ravine by running down, crossing the creek at a low level, and running up again on the other side. Even with two per cent. grades this lengthened the line some four miles, and it seemed better engineering to build a high-level viaduct than to avoid it.

An application was made by the late Gen. Thomas L. Kane, of Pennsylvania, president of the branch line, to the engineering firm of Clarke, Reeves & Co., of Phœnixville, Pa., to ascertain the cost of a high-level viaduct, and whether it would be possible to get this viaduct ready for use in due season. It should be remembered that at that time, 1882, the highest viaduct in existence, the Verugas viaduct in Peru, was 252 feet high, and this crossing required a viaduct of over 300 feet high.

Mr. Adolphus Bonzano, of Clarke, Reeves & Co., member of the American Society of Civil Engineers, told General Kane he could build him a viaduct 1,000 feet high if he would furnish the money, and then made a preliminary sketch, embodying those proportions and details which had been previously worked out on smaller structures.

Before going farther, it may be well to give a brief sketch of the origin of American railroad viaducts. Viaducts of no great height had been made of cast-iron columns both in the United States and in Europe, when a new departure was made by the late C. Shaler Smith, member of the American Society of Civil Engineers, by the use of girders of wood or iron, resting on columns of rolled iron sections, preferably the well known Phœnix column. He generally placed these columns thirty feet apart in the direction of the line of railway, and gave them a very considerable batir, not less than two inches per foot crosswise.

They were braced together longitudinally by horizontal struts of wood, in order to avoid the difficulties arising from the expansion and contraction of continuous lines of iron which could have no sliding motion.

The undersigned designed a viaduct approach for the proposed bridge across Blackwell's Island, N. Y., in 1870, for the then chief engineer, I. D. Coleman, Mem. Am. Soc. Civ. Engs.

Not wishing to use any perishable material like wood, the undersigned avoided the difficulties arising from changes of length due to changes of temperature, by simply leaving out the longitudinal struts and bracing of each alternate bay. This gave a series of braced towers thirty feet long, connected by girders of the same length resting on their tops, and having a sliding motion at one end.

Afterward it was found expedient to lengthen these intermediate girder spans to the greatest length which could be transported whole from the shops to the site, say sixty feet. (See drawing No. 1.)

This is believed to be the historical origin of the well-known American railroad viaduct.

English engineers have paid us the compliment of copying our viaducts in a late work on the Antifagasta railway in Bolivia, South America, called the "Loa viaduct."

The section of the columns is the same, and the dimensions are very closely followed, as may be seen from the following comparison:

	Kinzua.	Loa.
Length	2,050 ft.	800 ft.
Height	302 "	336½ "
Width of towers	38½ "	32 "
Connecting spans	61 "	80 "

	Kinzua.	Loa.
Depth of girders	6 ft.	8 ft.
Width of platform	18 "	13 "
Gauge of railway	4' 8½"	2' 6"
Inclination of pier posts	1 in 6	1 in 6
Wind pressure per sq. ft.	30 lbs.	30 lbs.
Weight of iron	1,400 tons.	1,115 tons.
Time taken to erect	4 months.	8 months.

A photograph of the Loa viaduct might easily be mistaken for the Kinzua.

In a structure of the height of the Kinzua viaduct, the question of wind pressure is a very serious one. American engineers had always recognized this in their practice, possibly from living in a country where cyclones rage. European engineers had not paid so much attention to this, until the destruction of the Tay bridge in Scotland forced it upon their notice very strongly.

After that catastrophe, the British Board of Trade fixed the amount to be provided for at 56 lbs. per square foot. Corresponding French practice was 300 kilos. per *centimètre carré*, or 52 lbs. per square foot.

The then chief engineer of the Erie railway, Mr. O. Chanute, member Am. Soc. C. E., considered that when the structure was unloaded, a wind pressure of 50 lbs. per square foot should be provided for, but when the structure was loaded from end to to end with a freight train, a pressure of 30 lbs. per square foot, including the area of the train, would be enough, as this would blow the train off the bridge.

When loaded, there is no upward pull at the base of the columns. When unloaded, there is an upward pull of six tons per column, which is resisted by anchor bolts, built into the pier masonry.

The live load provided for is that of a consolidation engine, weighing 80 tons, with 44 tons on its drivers in a space of 14 feet 9 inches, and followed by an uniform load of 1½ tons per lineal foot.

This viaduct was designed to be erected without scaffolding in as short a time as possible. It was erected in 4 months by a gang of about 40 men. This is at a rate of 500 feet per month, while the South American viaduct, above referred to, was erected at a rate of but 100 feet per month. The iron work was delivered at one end only of the ravine and slid down along a trough of

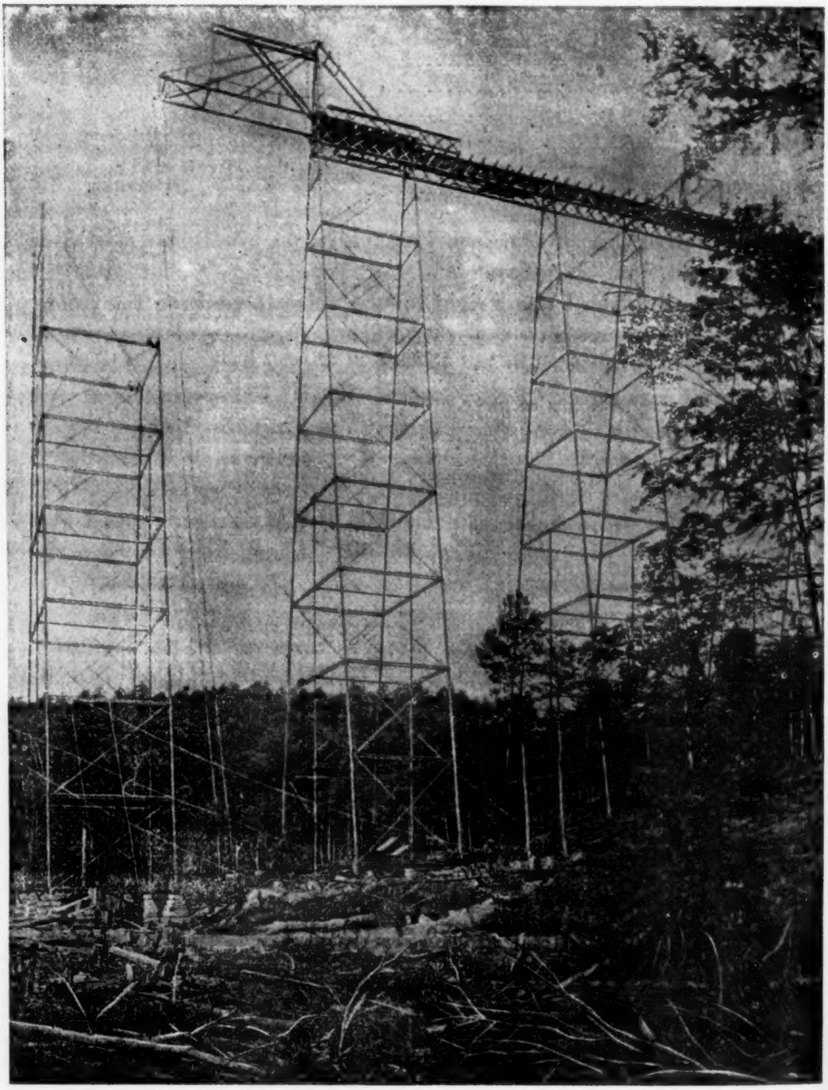


FIG. 199.

timber to its proper pier, then raised by a crane standing on the end of the last completed span. (See Fig. 198.)

The girders were raised complete in one piece. The parts of the piers were raised by gin-poles lashed to the tops of

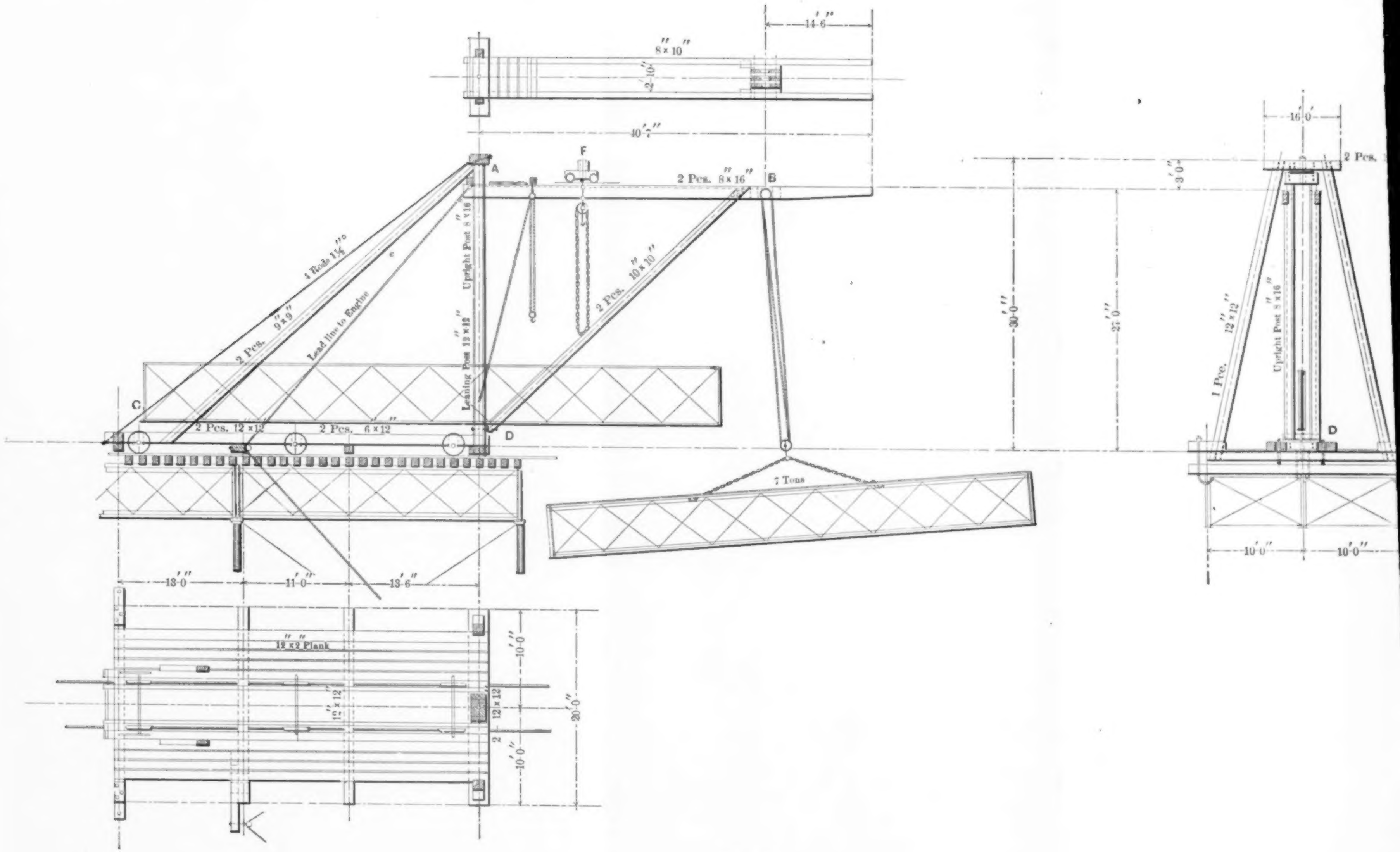

14' 6"
8" x 10"
2' 10"
40' 7"
F
A
2 Pcs. 8" x 16"
B
4 Rods 1¼"
2 Pcs. 9" x 9"
Lead line to Engine
Upright Post 8" x 16"
Leaning Post 12" x 12"
2 Pcs. 10" x 10"
30' 0"
27' 0"
3' 0"
C
2 Pcs. 12" x 12"
2 Pcs. 6" x 12"
D
7 Tons
13' 0"
11' 0"
13' 6"
12" x 2" Plank
12" x 12"
10' 0"
20' 0"
16' 0"
2 Pcs.
1 Pce. 12" x 12"
Upright Post 8" x 16"
10' 0"
10' 0"

FIG. 198.

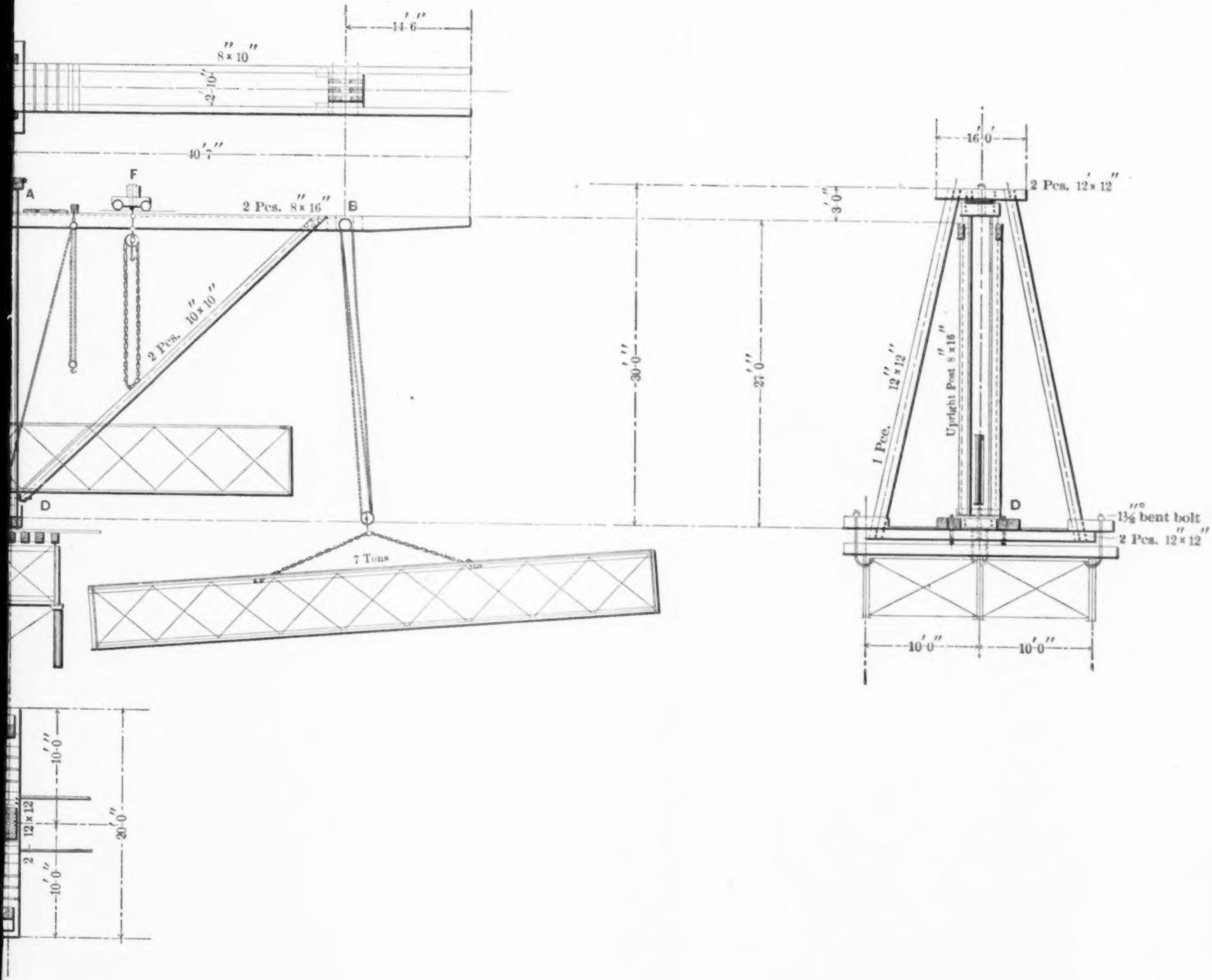

14' 6"
8"×10"
2' 10"
40' 7"
F
A
2 Pcs. 8"×16"
B
2 Pcs. 10"×10"
D
7 Tons
30' 0"
27' 0"
3' 0"
16' 0'
2 Pcs. 12"×12"
1 Pce. 12"×12"
Upright Post 8"×16"
D
1⅛" bent bolt
2 Pcs. 12"×12"
10' 0"
10' 0"
10' 0"
20' 0"
2 - 12×12
10' 0"

FIG. 198.

the columns, and raised up from story to story as the pier rose.

There was used about 3 miles of rope, and 2 hoisting engines built by Copeland & Bacon of New York, each having 4 spools that could be run separately.

The joints of the columns were at the junction of struts. A better detail would have been to have placed these joints a foot above the junction of struts, which would have expedited erection.

The whole erection was superintended by R. A. Simmons, superintendent of erection for Clarke, Reeves & Co.

The cost of erection did not exceed $12 per ton.

The following list of drawings accompany this paper:

Fig. 188. General elevation.
Fig. 189. Details of connections.
Fig. 190–195. Details of connections.
Fig. 196. Girders.
Fig. 197. Strain-sheet.
Fig. 198. Travelling crane used in erection.
Fig. 199. Copy of photograph showing process of erection.
Fig. 200. Specimen of column cut out in 1890, after 8 years' use.

NOTE.—Referring to photo-lithograph of piece cut from column of Kinzua Viaduct after eight years' service:

Copy.

NEW YORK, LAKE ERIE & WESTERN RAILROAD CO.,
NEW YORK. *March* 31, 1890.

C. W. BUCHHOLZ, Esq.,
Chief Engineer.

Dear Sir: In accordance with your instructions, I have cut from a column of Kinzua Viaduct the accompanying disk as a specimen, showing amount of corrosion which takes place on the inside of these columns.

In order that the specimen might show the worst conditions existing in the viaduct, one of the longest columns was selected and the disk taken from the under side. Any drip which might occur on the inside of the column would probably affect this point.

The specimen when taken from the column was entirely free from corrosion or any other sign of deterioration. The paint was in as good condition as when first put on, and where it was chipped off at the edges by the cutting tool, the iron showed a perfectly clean, new surface.

The specimen, $1\frac{3}{4}''$ in diameter and .989'' thick, was cut from the inside of the southwest column of the ninth bent from the east end of the viaduct, at a point about 3 ft. 6'' above the base of pedestal and about 12'' above the horizontal strut.

The date of cutting was March 29, 1890.

Respectfully yours,
HENRY B. SEAMAN.

The explanation of this is that the iron was protected by the paint from the action of the oxygen of the air. The paint itself was protected from the action of the sun, which on exterior surfaces first dries out the oil, and then, by causing a different rate of expansion and contraction between the paint and the iron, loosens the paint and leaves it to be washed off by rains.

T. C. C.

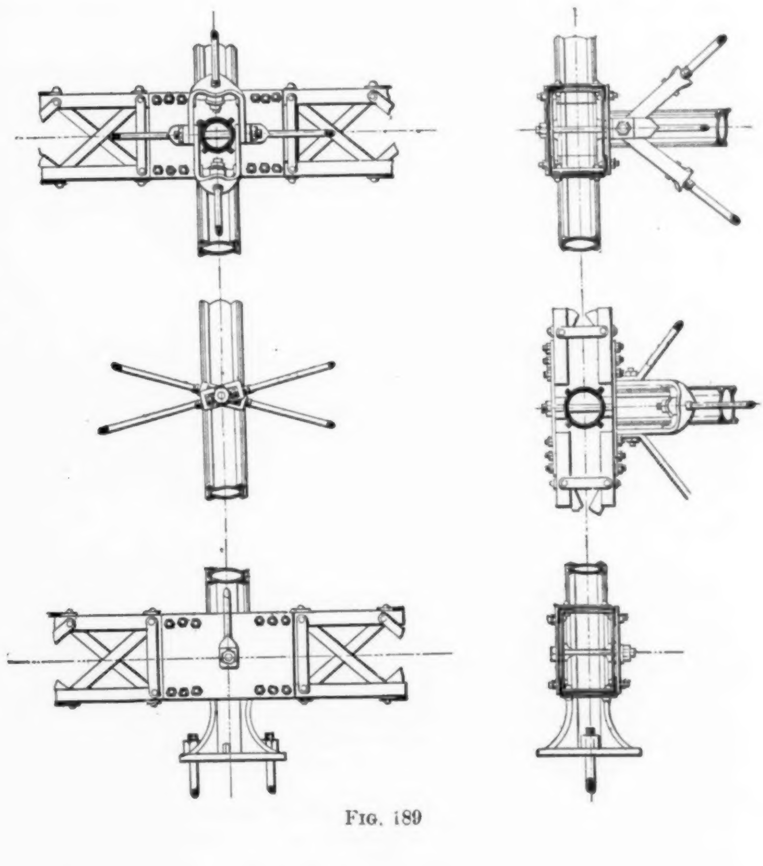


FIG. 189

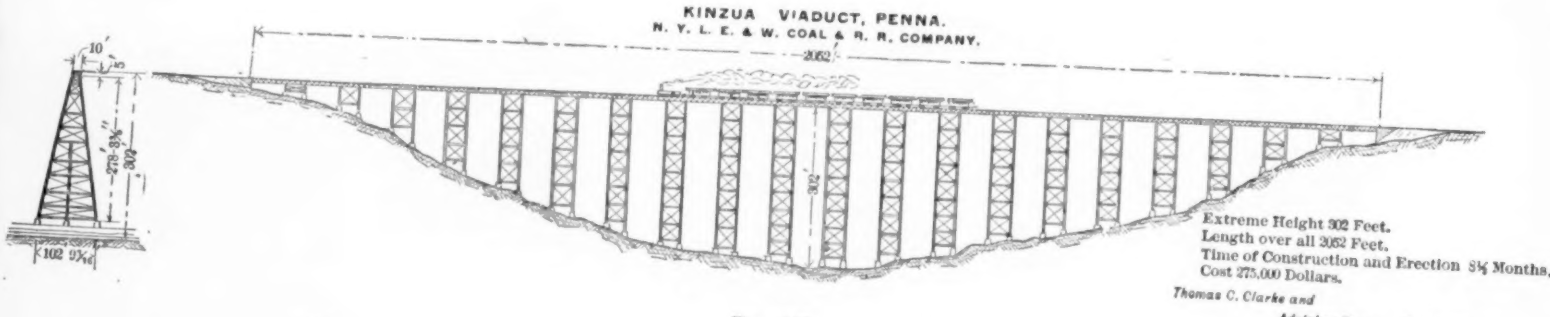


FIG. 188.

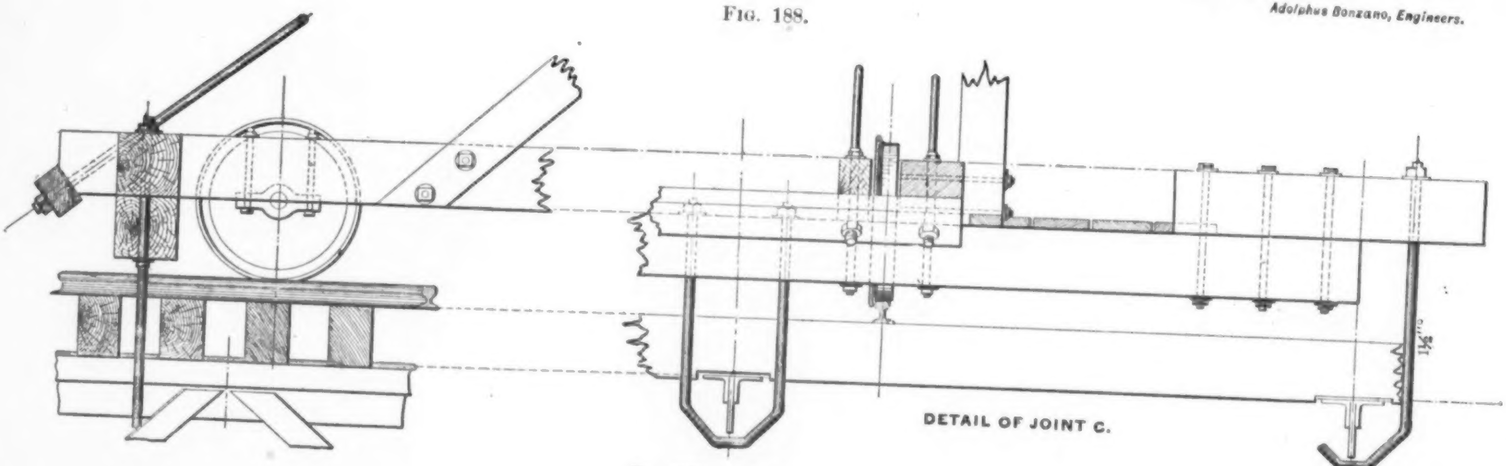


FIG. 192.—See Fig. 198.

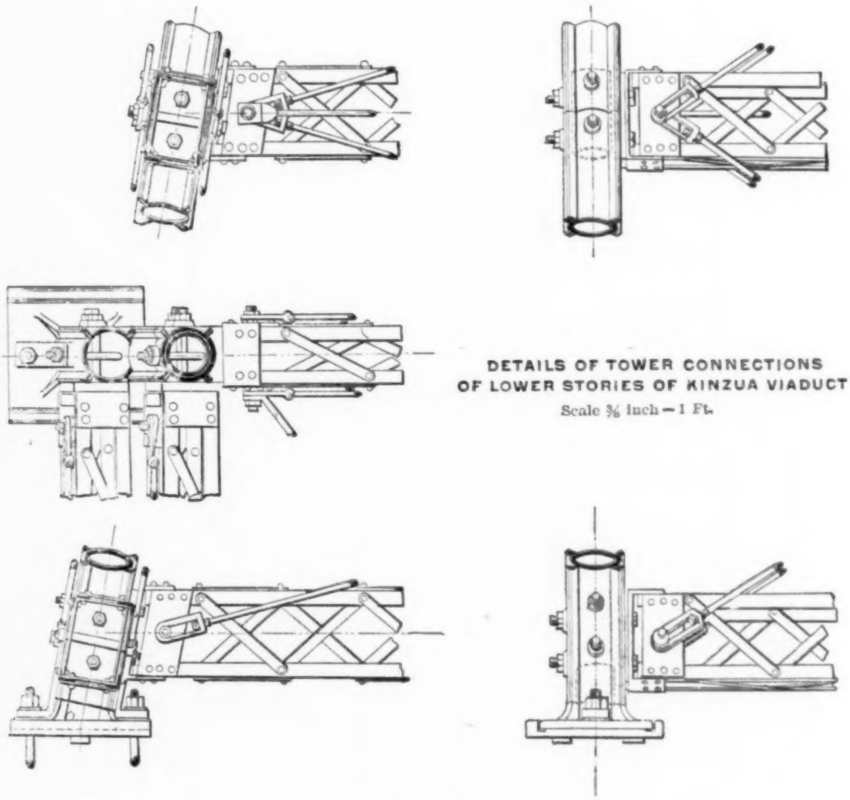


FIG. 190.

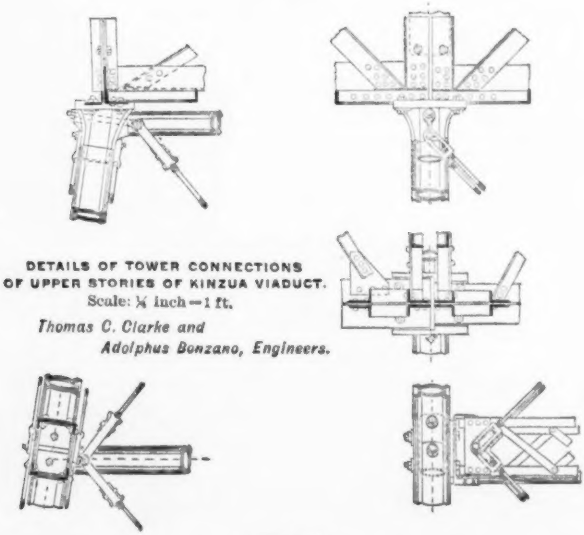


FIG. 191.

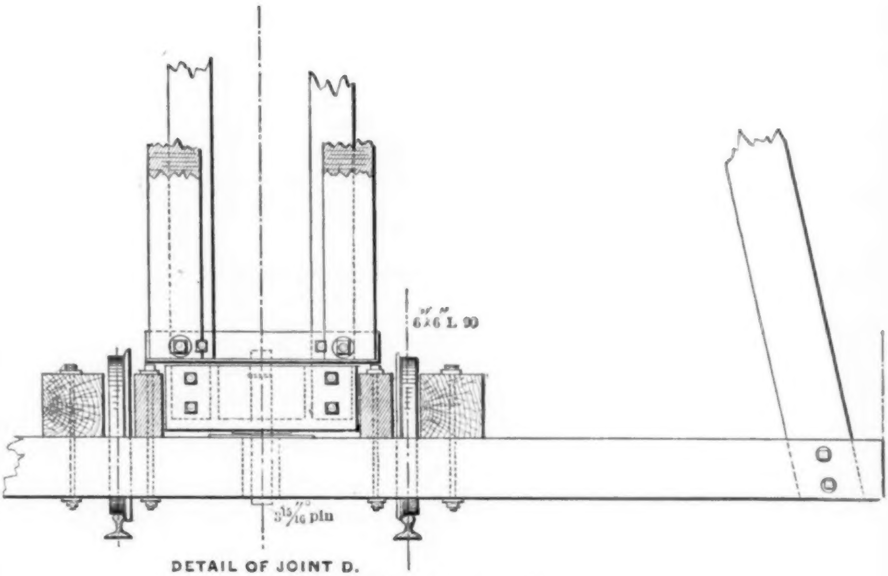


FIG. 193.—See Fig. 198.

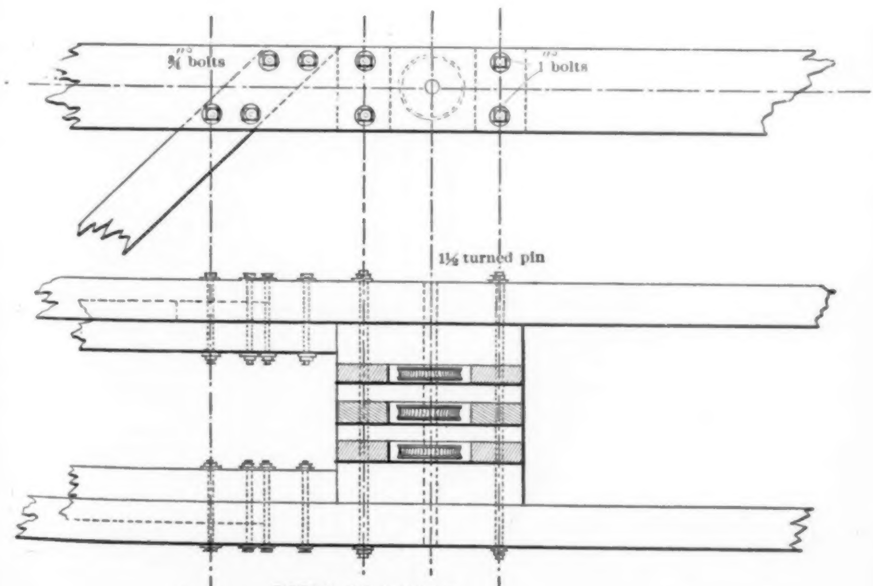


FIG. 194.—See Fig. 198.

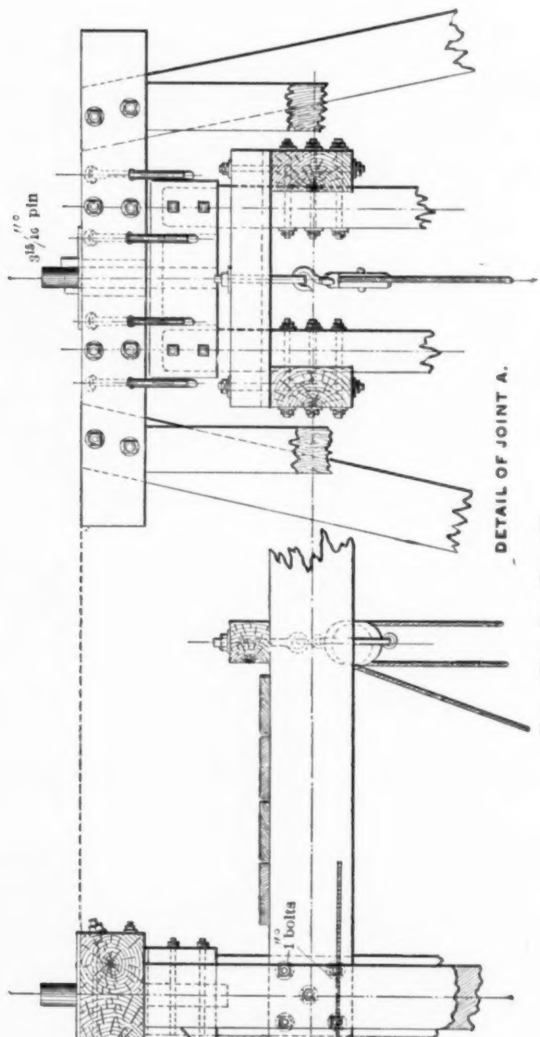
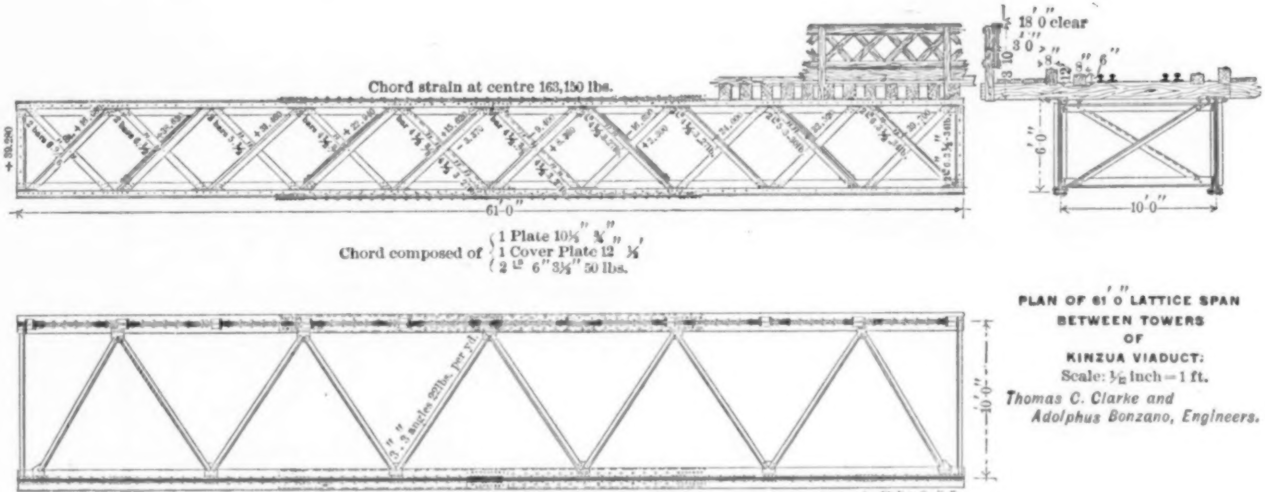


FIG. 195.—See Fig. 198.



DATA FOR STRESSES.

Live load: Consolidation engines having 88000 lbs. on drivers in a space of 14' 9"
Total weight of engine & tender 161,340 lbs. length over all 54' 3".
Dead load: 1000 lbs. per linear ft.

FIG. 196.

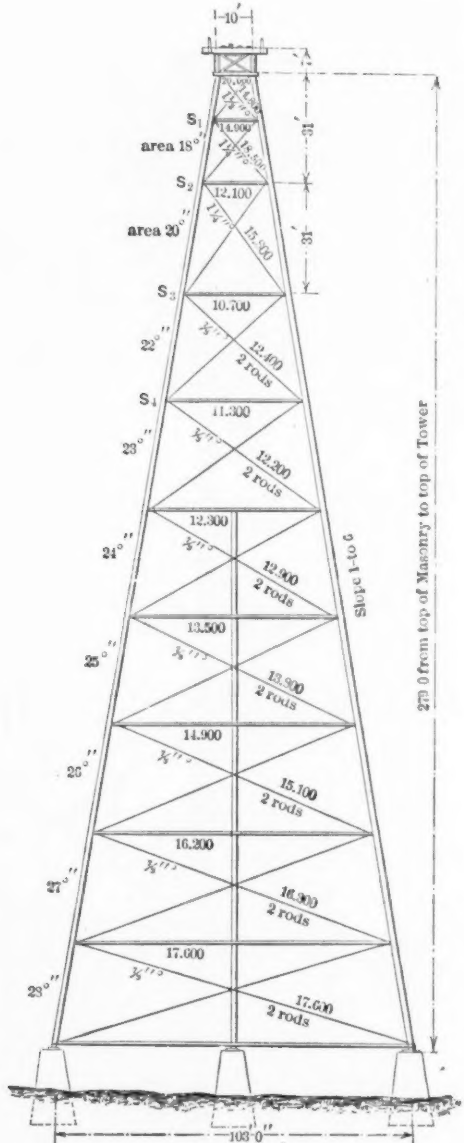
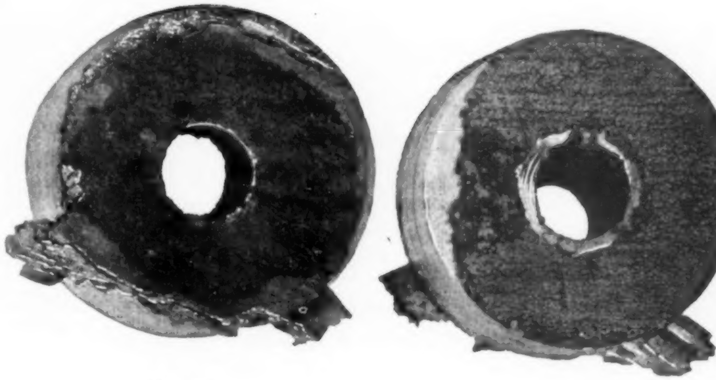

10′
S1
area 18″
14.900
S2
12.100
area 20″
15.800
31′
31′
S3
10.700
22″
12.400
2 rods
S4
11.300
23″
12.200
2 rods
12.300
24″
12.900
2 rods
Slope 1-to 6
279 0 from top of Masonry to top of Tower
13.500
25″
13.800
2 rods
14.900
26″
15.100
2 rods
16.200
27″
16.800
2 rods
17.600
28″
17.600
2 rods
103′0″
Scale 1″=50′

FIG. 197.



INSIDE. OUTS DE.

FIG. 200.

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CCCCI.

CHIMNEY DRAUGHT.

BY DE VOLSON WOOD, HOBOKEN, N. J.
(Member of the Society.)

SOME years since an author wrote: "It is pointed out by Peclet that the maximum draught occurs only with an infinite degree of temperature in the chimney, but that a temperature of about 600° Fahr. gives $\frac{7}{10}$ of the maximum.

"The statement in Rankine that the maximum quantity of air drawn in occurs for a temperature of the heated gases of about 600° Fahr. seems to be an error, as no such theoretical maximum exists in the formula for draught that has been given."

This quotation is given as a type of a class of criticisms which appear from time to time. It is occasionally asserted that actual experiments show that the maximum chimney draught does not occur at 600° Fahr., but that it is increased considerably by a considerable increase of temperature about this temperature. While it is not advisable to give much notice to casual observations in which measurements are not made, yet where a criticism is often repeated, apparently involving valid reasoning, as in this case, it is advisable to re-examine the theory to see if it admits of exceptions to the generally accepted results.

If the hypotheses of Peclet and of Rankine be admitted, there is no escaping the logic nor the conclusions reached by the analysis. Peclet's hypotheses are:

1. A certain amount of air must pass through the grate and the body of the coal on the grate to secure combustion.

2. Since the openings for the admission of air are fixed mechanically, the requisite amount of air must be supplied at a definite velocity.

3. That the required velocity may be produced by the pressure of a column of atmospheric air, which pressure will be the difference of the pressure of the external air and that within the furnace. The height of such a column is called a "head."

4. That, to the head described in the preceding condition, a head must be added sufficient to overcome the resistance offered by the coal to the passage of the air through it; and another head for the resistance offered by the flues and chimney to the passage of the gases produced by combustion.

It is certain that no serious objection can be raised against these hypotheses, the questions being in regard to the *law* which shall represent these conditions. Peclet represented it by the equation:

$$h = \frac{u^2}{2g}\left(1 + G + \frac{fl}{m}\right), \quad . \quad . \quad . \quad . \quad (1)$$

in which

u is the required velocity of gases *in the chimney*,

G, a constant to represent the resistance to the passage of air through the coal,

l, the length of the flues and chimney,

m, the mean hydraulic depth, or the area of a cross-section divided by the perimeter,

f, a constant depending upon the nature of the surfaces over which the gases pass, whether smooth, or sooty and rough.

If now

A be the section of the chimney in square feet,

b, the diameter of the chimney if round, or one side if square,

H, the height of the chimney in feet,

$\tau_0 = 461°$ Fahr., absolute (temperature of melting ice),

τ_1, the temperature of the gases in the chimney,

V_0, the volume of air at the temperature 32° Fahr., supplied per pound of fuel burned on the grate,

w, the pounds of fuel burned *per second*,

n, the ratio of grate area to that of the chimney area,

S, the area of the grate;

then

$$wV_0\frac{\tau_1}{\tau_0} = uA = nuS; \quad . \quad . \quad . \quad . \quad . \quad (2)$$

consequently the pounds of fuel burned per square foot of grate *per hour* will be

$$3{,}600\,\frac{w}{S} = 3{,}600\,\frac{nu}{V_0}\cdot\frac{\tau_0}{\tau_1}. \quad . \quad . \quad . \quad . \quad (3)$$

Peclet found that when 20 to 24 pounds of coal is burned per hour the value of G is about 12, and f for sooty surfaces, $f = 0.012$. In my computations, further on, I have assumed 12 for G except when 16 pounds is burned, when I assumed 11. As I have neglected the length of the flues, I have considered $f = 0.015$. Then we have, from equation (1),

$$h = \frac{V_0}{2g}\left(\frac{w}{A}\right)^2\left(\frac{\tau_1}{\tau_0}\right)^2\left(13 + \frac{0.060bH}{A}\right). \quad . \quad . \quad . \quad (4)$$

From this the required head may be computed. The departures in practice from the assumptions above made are considered by Rankine to be unimportant.

Rankine's determination of the height of chimney was for the purpose of supplying the head h. His hypotheses are :

1. The gases in the chimney are uniformly hot ;
2. The gases move in parallel sections through the chimney ;
3. The density of the gases in the chimney is uniform, and does not differ sensibly from that of air at the same temperature and pressure ; in other words, it is assumed that the density varies with the temperature only, the variation of pressure being neglected in determining the density.
4. "The head producing the draught in the chimney is equivalent to the excess of the weight of a vertical column of cool air outside the chimney, and of the same height, above that of a vertical column of equal base of the hot gases within the chimney."
5. That the draught is a maximum when the *weight* of gases discharged is the greatest.

The fourth hypothesis is improperly defined, since the head is defined as a weight, whereas it is a height in feet. We would define it as such a height of hot gases as, if added to the column of gases in the chimney, would produce the same pressure at the furnace as a column of outside air, of the same area of base, and a height equal to that of the chimney.

If 24 pounds of air be supplied per pound of fuel the volume of the gaseous product will be $24 \times 12\frac{1}{2} = 300$ cubic feet (nearly), and the weight of one cubic foot will be $\frac{1}{300} = 0.0033$ of a pound at 32° Fahr., which, added to the weight of a cubic foot of air at 32° Fahr., gives $0.0807 + 0.0033 = 0.084$ of a pound ; and if τ_2 be

the temperature of the external air, we have at once from the fourth principle as amended,

$$h = \frac{\frac{\tau_0}{\tau_2}\left(0.0807\right)}{\frac{\tau_0}{\tau_1}\left(0.084\right)} H - H = \left(0.96\,\frac{\tau_1}{\tau_2} - 1\right) H, \quad . \quad . \quad . \quad (5)$$

which is the formula given by Rankine. From (4) and (5) we find

$$H = \frac{13\,\frac{V_0^2}{2g}\left(\frac{w}{n\delta}\right)^2\left(\frac{\tau_1}{\tau_0}\right)^2}{0.96\,\frac{\tau_1}{\tau_2} - 1 - \frac{0.06b}{2gA}\left(V_0^2\,\frac{w}{n\delta}\,\frac{\tau_1}{\tau_0}\right)^2} \quad . \quad . \quad . \quad . \quad (6)$$

This gives the height of chimney for burning w pounds of coal per second.

Equation (5) involves the first four of the above hypotheses. In a properly proportioned chimney, working properly, the departure from the assumed uniform temperature will not seriously affect the analysis. If the chimney be too large for the volume of gases carried the central portion of the stream of gases may exceed the mean. Under proper conditions the second hypothesis will also be approximately realized.

In regard to the third hypothesis, no account is made of a diminution of pressure due to the velocity of the gases. In a statical condition, the pressure within the chimney would be that of the external air, but in practice it will be less, and hence the gases in the chimney will be more rarefied than found by assuming the pressure constant, thus increasing, somewhat, the coefficient 0.96. On the other hand, if less than 24 pounds of air be supplied per pound of coal, this coefficient would be less than 0.96; but none of these considerations seriously affect the theory.

Equation (5) gives the available head but it does not determine the velocity. Any velocity less than $\sqrt{2gh}$ may be secured by regulating the passage of air, and this may be done by piling coal on to the grate, or by using a heavier bed of finer coal, or by closing the furnace doors, or by dampers. Equation (1) assumes that the transverse section of the stream of air and gases is constant from the entrance to the furnace to the top of the chimney. It does not take account of the volume of air admitted. If the

supply be cut off the formula would indicate the same velocity in the chimney. It assumes that the entire head is available for producing velocity and overcoming resistances. The conditions under which Peclet determined the constants in equation (1) should be recognized. We shall, in the absence of anything better, assume Peclet's law. Equation (2) gives an independent value of u dependent upon the weight of fuel burned.

If δ be the weight of a cubic foot of the gases in the chimney, and

N, the number of pounds of air required per pound of coal (about 24 pounds),

then will the weight of gases passed up the chimney be

$$2\delta V_o \frac{\tau_1}{\tau_o} \delta = 0.0807 Nw \text{ nearly};$$

$$\therefore \delta = \frac{0.0807 N\tau_o}{V_o \tau_1}. \qquad (7)$$

The weight per second will also be δ times the volume, or $Au\delta$; hence equations (1), (5), (6), give

$$Au\delta = \frac{0.0807 N\tau_o A \sqrt{2gH} \sqrt{0.96 \frac{\tau_1}{\tau_2} - 1}}{V_{o1}\left(1 + G + \frac{0.06bH}{A}\right)}. \qquad (8)$$

Observing that $N \div V_o$ will be constant, this expression will be a maximum for a given chimney when the function

$$\frac{\sqrt{0.96 \frac{\tau}{\tau_2} - 1}}{\tau\left(1 + G + \frac{0.06bH}{A}\right)}$$

is a maximum. If G be a variable the maximum cannot be found unless it be a known function of the temperature. Too little, however, is known of its value in special cases, and much less can a law of variation be assigned to it. If it be considered constant, as Peclet and Rankine have done, the function for a maximum reduces to

$$\frac{0.96\tau - \tau_2}{\tau},$$

which is the function considered by Rankine, and gives $\tau = 2\frac{1}{12}\tau_2$ when it is a maximum; and if the temperature of the external air be 60°, then will the temperature of the gases be 622° for discharging the maximum weight of gases.

We have seen that this result is not a fixed value, but departures from theory in practice do not affect the result largely. If the coefficient 0.96 were 0.94, then the temperature should be

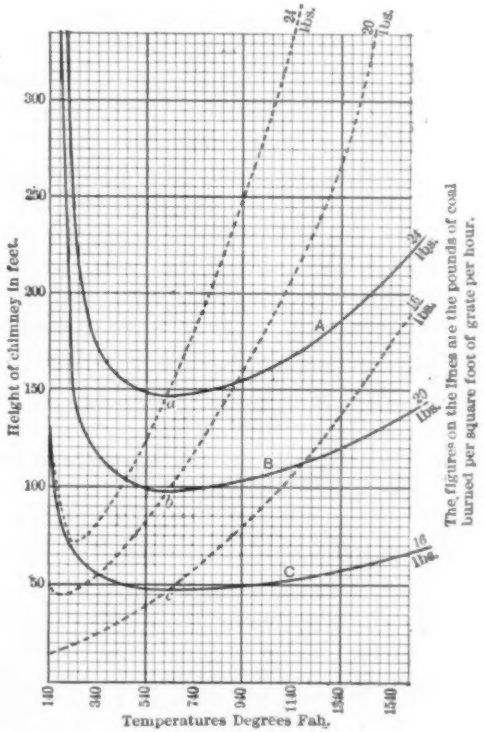

FIG. 201.

about 645°, and if it were 0.98, the temperature would be about 600°. There is then, in a properly constructed chimney, properly working, a temperature giving a maximum draught, and that temperature is not far from the value given by Rankine, although in special cases it may be 50 or 75 degrees more or less

Of the abnormal cases it is only necessary to consider that in which the section of the chimney is much too large and the height too small. In such a case not only may eddies be formed, but

the gases may rush through without involving the entire section of the chimney—very much like a flame from a pile of inflammable matter burning in the open air. This analysis is not applicable to such a case.

The foregoing diagram (Fig. 201) shows the law of relation between the height of chimney for three cases, marked *A*, *B*, *C*, in which it was assumed that 24 lbs. of coal were burned per hour per square foot of grate in case *A*, 20 lbs. for *B*, and 16 lbs. for *C*. These were computed from formula (6). The ordinates to the broken lines show the *heads* required in order to burn the 24, 20, 16 pounds for the respective cases, and were computed from formula (5). It will be seen that the required heads equal the heights of chimney when the temperature is about 620° Fahr., as shown at the points *a*, *b*, *c*; and for temperatures below 600° the height of chimney exceeds the required head, while for higher temperatures the required head exceeds the height of chimney. Also *the head has a minimum value*, which for chimney *A* is for a temperature of about 250° Fahr. The temperature of the external air was assumed to be 60° Fahr. The following is the table from which the diagram was constructed :

TABLE

SHOWING THE HEIGHTS OF CHIMNEY AND CORRESPONDING HEADS FOR BURNING GIVEN AMOUNTS OF COAL.

τ_2	τ_1 Absolute.	24 lbs. coal per sq. ft. grate area.		20 lbs. coal per sq. ft. grate area.		16 lbs. coal per sq. ft. grate area.	
		Head, h ft.	Height, H ft.	Head, h ft.	Height, H ft.	Head, h ft.	Height, H ft.
520°	600	128.061	1207.57	52.76	497.73	14.02	132.33
Absolute	700	72.75	250.87	45.70	157.61	19.65	67.76
or	800	82.76	172.42	55.58	115.80	26.71	55.66
59° Fahr.	1000	125.22	149.08	83.97	99.97	40.89	48.68
	1100	158.22	148.76	101.87	98.91	49.61	48.17
	1200	182.34	151.97	121.03	100.86	58.87	49.06
	1400	252.51	159.86	161.92	105.65	80.89	51.20
	1600	334.38	168.83	219.68	110.95	105.96	53.52
	2000	555.53	206.52	355.61	132.20	169.47	63.01

DISCUSSION.

Prof. H. B. Gale.—This paper gives a very clear exposition of Rankine's theory in regard to the temperature of chimney required to give maximum draught. In a paper presented at

the last meeting, I called attention to the apparent lack of agreement between the results of that theory and the results of observation upon actual chimneys, and I also there tried to point out briefly the main reasons for that disagreement. As the subject has excited some controversy, I may perhaps be allowed to restate those reasons from the different point of view taken in the present paper.

In the author's discussion of the conditions under which his expression for weight of gas discharged will be a maximum, he says: "If G be a variable, the maximum cannot be found unless it be a known function of the temperature."

To obtain the results deduced by Peclet and Rankine, G must be considered as constant; but it is not difficult to show that G is really a variable, and, moreover, a known function of the chimney temperature. It may be shown, also, that the value of G depends not only upon the temperature of the chimney, but upon various other conditions, notably upon the ratio of the chimney area to the area of the grate.

First, as to the effect upon the value of G of variation in the temperature of the chimney. G is a factor which, multiplied by $\frac{u^2}{2g}$, is to represent the resistance encountered by the air in its passage between the grate bars and through the coal. Now, with a given velocity of the chimney gases, u, the quantity discharged per second will be inversely as the temperature, T_1; and as the density of the air passing through the grate is not affected by variations in the chimney temperature, the velocity through the grate will be also inversely as T_1. Now, with a given grate and fuel-bed and a given temperature for the entering air, the resistance encountered in passing through the grate will be proportional roughly to the square of the velocity at that point. It follows that for a given velocity in the chimney, u, and other conditions remaining unchanged, the resistance of the grate will be inversely as the square of the chimney temperature. In other words, G varies inversely as the square of T_1. In order to practically maintain the velocity u constant, as supposed above, it is of course necessary, when the temperature in the chimney is increased, to close the dámper partly, thus interposing sufficient extra resistance to make up for the diminution of the resistance in the furnace.

Second, with a given velocity in the chimney and other con-

ditions constant, the velocity of the air passing through the grate will be proportional roughly to the ratio of the chimney area to the area of the openings through the grate. The area of opening for admission of air through the grate may be represented by a. We should then make G also proportional to the square of the ratio $\frac{A}{a}$.

The value of G would vary more or less with other conditions, such as fineness of fuel, thickness of fuel-bed, temperature of entering air, etc., but its law of variation with chimney temperature is of most importance in determining the value of that temperature which will make the draught a maximum. To ascertain that value approximately, we should therefore insert instead of G an expression of the form—

$$\frac{C}{T_1^2}\left(\frac{A}{a}\right)^2.$$

When it is observed that G is much the largest of the three terms in the parentheses in equation 8, so that the weight of gas discharged is nearly inversely as G, it is evident that inserting the proper value for this term would make a very considerable change in the expression for the temperature of maximum draught.

That expression is worked out with due regard to the principles here stated, and by what seems to me a simpler and more direct method, in the paper on chimneys presented last fall at the New York meeting. The temperature of maximum draught is there shown to be variable and to depend upon the proportions of chimney and grate, its value under ordinary circumstances being above 1,000° Fahr.

*Prof. De Volson Wood.**—Mr. Gale remarks that "the value of G depends not only upon the temperature of the chimney, but upon various other conditions, notably upon the ratio of the chimney area to the area of the grate."

It should be noted that the analysis is for a *given* chimney, and as the dimensions of the chimney and grate are supposed to be constant during use, G will not vary in a given case from these causes. Admitting that it may be true that the numerical value of G will depend upon this ratio, it nevertheless follows

* Author's closure, under the Rules.

that it will be constant for a given case so far as these dimensions are concerned.

There appears to be a misapprehension in regard to the essential part of the resistances represented by the symbol G. A part of it is of the nature of a frictional resistance, but this is the smaller part. The other part is equivalent to closing a part of the grate opening, and this is the greater part. Conceive that the temperature in the chimney is increased, then, in order to maintain a constant velocity in the chimney, the resistance in the furnace must be increased—not decreased, as stated by Prof. Gale. The desired result may be secured by partly closing the openings into the furnace or by adding more fuel. The effect of the latter will be not only to partly close the openings before existing, but add somewhat to the lengths of the passages through the coal. Neglecting the effect of the increased depth, the two modes of diminishing the supply of air will be substantially the same. It would, then, be more nearly correct to say that—other things being the same—the openings for the supply of air should be reduced in proportion to the absolute temperatures in the chimney. But the frictional resistances are very nearly independent of the size of the openings and vary with the square of the velocity. We conclude, therefore, that *for a given bed of coal* in a given furnace the value of G is practically constant.

CCCCII.

TESTS OF RECENT FORMULÆ FOR CHIMNEY DRAUGHT.

BY WILLIAM KENT, M.E., NEW YORK.

(Member of the Society.)

SOME years ago, while endeavoring to make a table of chimney dimensions for different horse-powers of boilers for practical use by the boiler company with which I was then connected, I took a number of published formulæ and rules for dimensioning chimneys, and calculated arithmetically the heights and diameters they would give for certain supposed cases within the limits of ordinary practice. I also took the dimensions of actual chimneys which had in practice given apparently good results, and plotted them all on cross-section paper. After considerable study of the results thus obtained I framed the empirical formula published in the Transactions, Vol. VI., page 81, and calculated from it the table appended to the paper. This table has been reprinted quite extensively in trade literature and elsewhere, and has been tested in practice by several engineers, to my knowledge with good results. I have not, as yet, seen any reason to modify either the formula or the table.

I have made a similar arithmetical calculation of results which may be obtained from the formulæ given in a recent paper before this Society,* which may prove interesting.

The author, in this paper—read at the New York meeting—gives a formula for "the most economical area" of a round chimney which is to burn F pounds of coal per hour, as follows:

$$\text{Formula (20), } A = .07F^{\frac{1}{2}}$$

On the same page he says: "A sufficient approach to the

* Horace B. Gale, Theory and Design of Chimneys, Transactions, Vol. XI., p 451.

most economical value can usually be attained by the handy rule, that the sectional area of the chimney in square feet should be equal to the number of pounds of coal to be burned per minute."

Assuming that a chimney designed for a boiler of a certain rated horse-power should be large enough to cause the burning under the boiler of 5 lbs. of coal per rated horse-power, the following results are obtained for boilers of 20, 40, 60, 80, 100, 200, 400, 600, and 1,000 H.P.:

Rating of Boiler, H.P.	Coal burned per Hour, F.	Coal burned per Minute, Area of Chimney by "Handy Rule."	Area by Formula, $A = .07F^{\frac{3}{4}}$	Ratio. $\frac{F}{A'}$	Ratio. $\frac{F}{A}$ nearly.
		A', Pounds or Square Feet.	Square Feet.		
20	100 lbs.	1.67	2.212	60	45
40	200 "	3.33	3.724	60	54
60	300 "	5.	5.096	60	59
80	400 "	6.67	6.258	60	64
100	500 "	8.33	7.399	60	68
200	1,000 "	16.67	12.460	60	80
400	2,000 "	33.33	20.937	60	95
600	3,000 "	50	28.378	60	106
1000	5,000 "	83.33	41.622	60	120

Showing that for 60 H.P. the handy rule and the formula give about the same result, while for chimneys less than 60 H.P. the "handy rule" gives the smaller area, and for chimneys larger than 60 H.P. it gives the larger area. In the case of 1,000 H.P. the handy rule gives twice as large an area as the formula.

Using the areas given by the formula, as having probably the greater approximation to accuracy, let us see what results are obtained by substituting them in the formula given by the same author for height of chimney.

The formula (16) of the paper, "A general equation for determining the height of a chimney of known cross-section which is to do a given work," after the substitution of C, the coefficient of friction of the gas on the sides of the stack, equal to .014, and B, the number of pounds of air supplied to burn a pound of fuel, equal to 21, becomes formula (17)

$$H = \frac{KT_s}{T_s - 533 \frac{M}{A^3}\left(\frac{T_sF}{15000}\right)^2} \left(\frac{F}{3.5a}\right)^2;$$

in which :

H = Height of chimney.

T = Mean absolute temperature F of gas in the stack.

M = Perimeter of the stack for round chimneys = $3.54 A^{\frac{1}{2}}$.

F = Number of pounds of fuel to be burned per hour.

a = Total area for admission of air in square feet.

A = Area of chimney in square feet.

K = Coefficient of resistance for the furnace.

The author says that in ordinary boiler furnaces the area a may be considered, roughly, as one-third of the grate area, and that when the ratio of grate surface to tube "calorimeter"* is between 6 and 9, K may be taken, approximately, as 0.2.

If the grate area is nine times the chimney area, then $a = 3A$.

Assuming that for average practice the temperature of the flue gases may be taken as 538.2° Fahr., or $T = 538.2° + 461.8° =$ 1,000° absolute, that $K = 0.2$, and that $a = 3A$; then, by substituting these values in formula (17), the value of H may be expressed in terms of A and F. Thus :

$$H = \frac{200}{467 - \dfrac{.0157 A^{\frac{1}{2}} F^2}{A^3}} \times \frac{F^2}{110.25 A^2}.$$

Using the values of A given by the formula (20), $A = .07 F^{\frac{1}{2}}$, in the table above given for boilers of 20, 60, 100, 200, 400, 600, and 1,000 H.P., we obtain the following:

H.P.	Fuel burned per Hour, F.	Area $A = .07 F^{\frac{1}{2}}$	Height in Feet, nearly.
20	100	2.21 sq. ft.	8
60	300	5.10 "	14
100	500	7.42 "	19
200	1,000	12.46 "	28
400	2,000	20.94 "	39
600	3,000	28.38 "	47
1,000	5,000	41.62 "	61

The tremendous difference between these figures for dimensions of chimneys and those generally used in practice, or which

*I put this word in quotation marks because I think it is an incorrect term as here used, although it has been used in this significance by many writers on steam engineering. Its proper meaning is an apparatus for measuring quantity of heat, as the "Barrus Calorimeter."

may be obtained by use of any generally accepted table or formula, suggests that either I have made a great mistake in my arithmetical calculation, or that there is something wrong with the formula which was used. If the former, I must then take exception to the formula in that it is so unhandy and liable to lead to arithmetical mistakes. If, however, the formulæ are correct, it is greatly to be desired that, by means of them, table may be computed which will be as handy as the one given in the *Transactions*, Vol. VI., page 83.

DISCUSSION.

Prof. H. B. Gale.—Two alternatives are suggested in this paper to account for the difference between the heights calculated by formula for a series of chimneys of given power, and the heights generally used in practice. The first is that perhaps the author has made a mistake in his arithmetical work. In regard to this I can say that I have gone over the calculations with some care, and have been unable to find any mistake. I may say also, in passing, that Mr. Kent would have found it easier, if, instead of using the rather cumbrous general formula which he has chosen, he had used the formula recommended for designing boiler chimneys in the paper referred to (viz. Equation 22, p. 464), which makes the height,

$$H = 100 \frac{K}{t} \left(\frac{F}{a}\right)^2,$$

where K is an experimental coefficient of friction, t is the chimney temperature on the ordinary Fahrenheit scale, F the number of pounds of coal to be burned per hour, and a the area of opening in the grate. This formula is not at all "unhandy," and gives practically the same results as the more general one which he has used.

Laying aside, therefore, the first alternative suggested by the author, and granting, for the sake of the argument, that the ordinary practice in regard to chimneys is the best possible, admitting also that the heights given in the paper do not agree with the ordinary practice, shall we accept his second alternative, and conclude that the formula, and the theory on which it is based, are wrong? The author has not offered to point out

wherein the theory is wrong; and there is another alternative, which may have escaped his notice, but which fully accounts for the disagreement.

It is well known that the height of the chimney is one of the main factors in determining the *rate of combustion* attainable in a boiler furnace. Changing the height of the chimney will affect in a marked degree the number of pounds of coal burned per square foot of grate. Conversely, if we calculate a chimney for a rate of combustion below the ordinary rate, we ought, if our formula is correct, to get a height also considerably less than the ordinary height.

Mr. Kent, in his calculations, has assumed the area of opening through the grate equal to three times the area of the chimney; which for an average grate, having an opening equal to three-eighths of its surface, would correspond to making the ratio of grate surface to chimney area about eight to one. It is a common practice to make the area of the chimney one-eighth that of the grate, and for chimneys of very large power this is doubtless an excellent proportion. The formula which I have recommended, however, gives a greater proportional area to chimneys of small power, allowing for their greater frictional resistance. Now, the author of the paper before us, by using my formula for chimney area, and still keeping the grate surface eight times the area of the chimney, has taken areas of grate for the small powers out of all proportion to the amounts of coal to be burned on them—in other words, he has calculated heights for chimneys suitable to inordinately *low rates of combustion.*

To ascertain heights suitable to the average conditions of practice, he should have assumed the area of grate opening, a, equal to about three-eighths, or 0.4, that of the grate, and then taken the area of the grate *such as is customary* for a boiler of the given power, or such as would give a rate of combustion corresponding to those generally employed.

When conditions are assumed so far away from the average practice as to allow over 16 square feet of grate to a 20 H.P. boiler, with a rate of combustion of only $5\frac{3}{5}$ lbs. of coal per hour per square foot of grate—as has been done in these calculations—and when, at the same time, a value of the friction factor, K, is employed, which has been given as adapted to "ordinary boiler furnaces," there is no reason to conclude that the formula is wrong because it gives a height of chimney considerably less

than the usual practice. There would be better ground for such a conclusion *if it did not.*

If it be desired to make a fair comparison between the heights of chimneys commonly used with boilers of various power, and the heights given by the formula which I have proposed, conditions must be taken to correspond with ordinary practice in the matter of grate area and rate of combustion as well as in other particulars. As it may be interesting to see how the results of the formula do compare with practice, *when the ordinary conditions of practice are assumed in the calculation,* I present here a table of heights calculated for the same powers and same quantities of coal burned as those given by Mr. Kent, temperature in the chimney being taken as 500°, and the number of pounds of coal burned per hour per square foot of grate ranging from 13 to 25 lbs. Rates above and below these limits are occasionally found, but it will, I think, be admitted that this range fairly represents ordinary steam-boiler practice in the United States.

H. P.	Coal per hour, F.	Coal per sq. ft. grate, R.	Area grate, G. $\left(=\frac{F}{R}\right)$	Area opening, a. $(=0.4\,G.)$	Height, $H=100\,\frac{K}{t}\left(\frac{F}{a}\right)^2$
	lbs.	lbs.	sq. ft.	sq. ft.	ft.
20	100	13	7.7	3	44
60	300	15	20	8	56
100	500	17	30	12	70
200	1000	19	53	21	90
400	2000	21	95	38	111
600	3000	23	130	52	133
1000	5000	25	200	80	156

Using the values in the third column for the rates of combustion, and taking the area of grate opening as 0.4 of the total grate area, the formula gives heights ranging from 44 feet for the 20 H.P. boiler to 156 feet for the 1,000 H.P. These heights, I think, agree fairly well with ordinary practice; at least, the heights given by this formula, *when applied to average conditions,* agree very well with those recommended by Mr. Kent's table, in Vol. VI. of the *Transactions,* for boilers of the same power.

I would like to say here that I consider Mr. Kent's table an excellent one, as giving proportions for boiler chimneys which

will work well in practice, whenever *the conditions* are such as correspond fairly with what may be called the *average conditions.* His formula and table, being derived, as he has told us, empirically, by a comparison of the results of ordinary practice, ought certainly to apply to those conditions. Empirical formulas, however, are usually unsafe when applied to conditions different from those on which they are based. Mr. Kent's formula makes no allowance for variations in the grate area, or in the per cent. of grate opening, and no allowance for variation in the chimney temperature; accordingly, we should not expect such a formula to give uniformly correct results for conditions which depart from the average in these respects. Such departures, however, are by no means uncommon, are often desirable, and sometimes necessary; and it is in the attempt to meet such variations in the conditions that my chimney formulas are offered.

As has been shown, when the conditions correspond with the average conditions of practice, they give heights substantially the same as are usually employed; when the conditions are different, they give different results, as should be expected.

The rates of combustion and grate-areas assumed in the calculations presented in the paper before us are evidently such that the resulting heights cannot fairly be compared with "those generally used in practice;" but the statement that there is a "tremendous difference" between the figures given in the paper and those which may be obtained by *any generally accepted formula*—applied to the *same conditions*—will not, I think, bear examination.

Formulas for chimneys are so various that there is room for some disagreement, perhaps, as to what is a "generally accepted" one. Probably the formulas given by Peclet and Rankine are as generally accepted as any. I have entered my objections to those formulas for general application, in that they make no account of variation in the ratio of grate to chimney area. However, for a ratio of 8 to 1, and temperatures from 500° to 600°—conditions similar to those for which Peclet's constants were determined—they may be safely used.

The equations given by Rankine in *The Steam Engine* may be reduced to the following form, by which may be calculated the number of pounds of coal that may be burned per hour per square foot of grate with a chimney of given height, and a ratio of grate to chimney of 8 to 1.

$$R = \frac{F}{8A} = \frac{450}{V_0} \frac{\tau_0}{\tau_1} \sqrt{\frac{2gH\left(.96 \frac{\tau_1}{\tau_2} - 1\right)}{13 + \frac{.012H}{m}}}$$

In this formula, using the same assumptions as are made in Mr. Kent's calculation, we have :

Absolute temperature of chimney, τ_1, $= 1000°$
For a round chimney, the hydraulic mean depth, m, $= .28\sqrt{A}$
For 21 lbs. of air per lb. of coal, the volume at temperature τ_0, or V_0, $= 262$
Absolute temperature of external air, τ_2, $= 520°$
$\tau_0 = 493°$

Another formula which has been quite widely published, which applies to about the same conditions as are assumed in Mr. Kent's calculation, and which agrees well with Isherwood's experiments on anthracite coal, is given by Professor Thurston.* This rule is, *Subtract one from twice the square root of the height, and the result is the rate of combustion for anthracite.* For low-grade soft coals the result is to be multiplied by 1.5. For the general case, we may use the mean of the figures for best anthracite and low-grade bituminous, or a multiplier of 1.25.

In the following table I have given in the second column the rates of combustion which the calculations in the paper before us show, on the assumed ratio of grate to chimney of 8 to 1, and in the third column the heights of chimney there deduced, by my formula ; in the last two columns are the rates of combustion, which the same chimneys should give, under the same conditions, according to the formulas of Rankine and of Professor Thurston.

H. P.	Coal per sq. ft. grate (assumed in paper), or $\frac{F}{8A}$.	Calculated height. H.	Coal per sq. ft. grate for height H.	
			Rankine.	Thurston. $F = 1.25\,(2\sqrt{H} - 1)$.
	lbs.	ft.	lbs.	lbs.
20	5.65	8	4.86	5.82
60	7.35	14	6.42	8.10
100	8.42	19	7.47	9.65
200	10.	28	9.06	11.98
400	11.9	39	10.7	14.4
600	13.2	47	11.7	15.9
1000	15.	61	13.3	18.3

* *Manual of Steam Boilers*, 1888, p. 324.

It will be seen that the results by my formula differ from the others, in each case, less than the two latter differ from each other.

A comparison of one formula with another, or even with average practice, is not a very satisfactory or accurate test of the correctness of the formula, as the accuracy of the standard is too uncertain; but there seems to be hardly enough difference in this case to be fairly characterized by the adjective tremendous.

I have generally obtained most satisfaction by comparing the formulas with the results of carefully made experiments upon chimneys, where all the conditions affecting the problem have been determined as accurately as possible. The results of one such comparison are given in my paper presented at the last meeting. Another, which may be of interest, is furnished by the results of a boiler trial made April 5 at the Power Station of the Missouri Railroad Company in St. Louis. The data are as follows: height of chimney, $H = 92$ feet; diameter, 50 inches; area, 13.12 square feet; area of grate surface, 48 square feet; area of grate opening, 21 square feet; temperature of chimney gases, 609° Fahr. The proportions of the boiler and furnace were such that the value $K = 0.2$ may be safely used. The number of pounds of coal burned per hour by the formula should then be, $F = 1{,}112$. The coal used was a rather free-burning bituminous coal, and the number of pounds per hour actually burned during the trial was 1,179. The damper and ash-pit doors were kept wide open all the time. According to the formula given by Mr. Kent in Vol. VI. of the *Transactions*, the coal consumed in this case ought to have been 1,749 lbs; but it is fair to call attention to the fact that as the ratio of grate to chimney in this case was not 8 to 1, this is one of the numerous cases in which the conditions are outside the limits within which the empirical formula referred to is safely applicable.

In regard to the first table presented in Mr. Kent's paper, the disagreement shown between the formula and the handy rule in the matter of the area of the larger sizes of chimneys illustrates the recommendation made in my recent paper that for the larger chimneys the formula should be used, as giving the most economical dimensions.

Mr. A. F. Nagle.—Some years ago, when I was engaged in mill

engineering, I tried to find a satisfactory formula for the size of chimneys.

They all seemed to me to be too complex for safe guidance. I could not help regarding the height as an element too greatly dependent upon conditions likely to vary so much as to make any scientific determination of the height really unsafe to use, other than as an approximate result at the best. I was about ready to say, Make chimneys as high as you are willing to pay for, but I concluded to ignore the height as an element other than to make it reasonably proportionate to the diameter. The coal burned seemed to me to be the safest basis for the area of the chimney, and I examined a good many chimneys, varying from 300 to 1,000 H.P., to see if there existed any common ratio between these two elements. Where a smaller ratio than one and one-half ($1\frac{1}{2}$) square inches of area to one (1) pound of coal burned per hour existed, much dissatisfaction existed as to the draft of the chimney. When it reached two (2) square inches per one pound of coal very satisfactory results were found to exist. Chimneys in full use rarely had more than this, although it is not unusual to make new chimneys so large as to permit doubling the boiler capacity. The handy rule quoted by Mr. Kent is equivalent to two and a quarter ($2\frac{1}{4}$) square inches of area per pound of coal per hour, and I should believe it to be very generous, and should give good results.

Prof. Gale.—I would say that that rule is only given as applicable to smaller sizes of chimney. In my paper, where I have given the formula, I have stated that it should be applied only to small sizes.

Mr. Scheffler.—I would like to ask Prof. Gale what he took as representing t in the formula in obtaining the figures meant for height.

Prof. Gale.—t equals 500° temperature on the ordinary Fahrenheit scale.

*Mr. Wm. Kent.**—Prof. Gale states as the reason why I obtained such unusual figures for height of chimney, in using his formula, that I have taken areas of grate for the small powers out of all proportion to the amounts of coal burned on them, or, in other words, that I have calculated heights for chimneys suitable to inordinately low rates of combustion. To ascertain heights suitable to the average conditions of practice, he says I

* Author's Closure, under the Rules.

should have assumed the area of grate opening a equal to about three-eighths, or 0.4, that of the grate, and then taken the area of grate such as is customary for the given power, or such as would give a rate of combustion corresponding to those generally employed.

Possibly I should have made these assumptions in using Prof. Gale's formula, but I thought it safer to assume the same values for these quantities which Mr. Gale himself uses or suggests in his own paper. In that paper he says: "In ordinary boiler furnaces the area for admission of air to the fire a may be considered roughly as one-third of the grate area, and when the ratio of grate surface to tube calorimeter is between 6 and 9, K may be taken approximately as 0.2." I therefore assumed in my calculations a to be equal to one-third of the grate area, the grate area to be 9 times the chimney area, and K to be 0.2, so as to make all my calculations as strictly within the limits of Prof. Gale's paper as possible.

Prof. Gale's direction that, in using his formula, I should have "taken the area of the grate *such as is customary* for a boiler of the given power, or such as would give a rate of combustion corresponding to those generally employed," is too indefinite to be followed in practice. What is the rate of combustion generally employed? I find that in the tests at the Centennial Exhibition, among the 5 boilers which gave the highest economical results out of the 14 reported on, in the economy test one boiler had a rate of combustion of 7.24 lbs. of anthracite per square foot of grate per hour; another boiler, not very different in construction and almost identical in economy, had a rate of 12.96 per square foot of grate. In the capacity tests these boilers used respectively 11.31 and 17.36 lbs. per square foot. In the table of 30 tests of Babcock and Wilcox boilers, published in *Steam*, the rate of combustion in those tests in which the boilers developed more than their rated horse-power varies from 11.21 to 40 lbs. per square foot of grate per hour. In a test which I made recently with a pair of tubular boilers on two consecutive days, the rate of combustion was 18.5 lbs. and 15.6 lbs. per square foot of grate. On both days the conditions were alike as far as they could possibly be made, damper and ash-pit doors wide open all day, but the coal on the first day gave only 8.4% of ashes and refuse, and the coal used on the second day 14%. The ashes and clinker gave so much trouble on the

second day that it was evident that it would have been much better to have had a larger grate surface and a still lower rate of combustion than 15 lbs. per square foot; while on the first day, no doubt, as good results as were obtained with 18.7 lbs. could have been obtained with a rate of 25 lbs. if the grate surface had been made smaller and the chimney increased in height so as to produce sufficient intensity of draught to burn coal at that rate.

In view of these differences in rates of combustion, which are to a large extent necessary, both on account of the extreme variations in character of coal, and of the variations in construction of boilers, which limit more or less the room available for grate surface, or which offer more or less obstruction to the passage of the gases from the fire to the chimney, it is manifestly impossible to give a figure for area of grate or rate of combustion "generally employed." If Prof. Gale had intended his formula to be used in practice he should have given numerical values to the coefficients to be used under various circumstances, instead of leaving to the judgment of each one who attempts to use the formula what rate of combustion or extent of grate surface is "generally employed."

Prof. Gale says I have not attempted to point out where his theory is wrong. I have purposely left that branch of the subject to others who discussed his paper at the New York meeting, and have confined myself to an attempt to apply his formula in practical cases.

I would, however, make the general criticism that it is wrong to construct a formula for use in practice in which the "constants" are derived by averaging a number of figures which are extremely variable, and many of them undeterminable, such as the free area of grate surface, the rate of combustion, the resistance of the fire grate and the bed of coals upon it, and other frictional resistances. These latter quantities vary enormously, and vary with every fresh firing and every time the fireman breaks up a coked mass on the surface of the fire or clears the clinkers from beneath it. An average of such variable quantities should not be taken as representing "customary conditions" or "ordinary practice." It is wrong to make a formula for dimensioning a structure which is to last for years depend upon such a quantity as the free area through the grate, which may vary with every variation in quality of coal used, and upon the re-

sistance to the passage of the air through the bed of coal—one of the variables entering into the coefficient K—which varies every minute of the day. The chimney, when built, is a fixed structure. It should be dimensioned with reference to the quantity of coal to be burned, which is the one quantity most likely to be approximately known in advance, and not to the free grate area or total grate area, which may be varied from time to time as the quality of coal may require, nor to any supposed coefficient of resistance to passage of air through the grates, which is in practice not determinable.

In the present state of our knowledge I do not think any satisfactory theory of chimneys can be framed which will include a consideration of all the different variables that affect the rate of combustion, and from which a formula can be derived that will prove of value in practice. Chimneys are not designed by engineers from Peclet's formulæ or from Rankine's, nor will they, I think, be designed from Prof. Gale's. The best chimney formula that can be obtained at present is an empirical one—which may be modified or divided into two or more to meet different practical conditions, as new data are obtained from experiment.

I regret that the principal debater of my paper has not seen fit to give us the dimensions which he would propose, calculated by the use of his formulæ, for chimneys for the several assumed cases which I figured upon, viz.: chimneys to be used with boilers consuming from 100 to 5,000 lbs. of coal per hour. If he had done so, we might have more easily discovered some practical value in the formulæ suggested.

CCCCIII.

THE GRAPHIC REPRESENTATION OF THERMAL QUANTITIES.

BY DE VOLSON WOOD, HOBOKEN, N. J.

(Member of the Society.)

THE geometrical representations of thermal quantities are of great assistance in forming mental conceptions of the changes which take place in fluids, due to changes in the heat to which they are subjected, and especially furnish a means of illustrating the algebraic expressions which occur in the study of thermodynamics.

We cannot say when graphics was first used in this science, but the use of thermal lines would naturally follow closely the discovery of Mariotte's law, $pv = a\ constant$, and areas would be used to illustrate propositions in the dynamic theory of heat.

The writer, so far as he is aware, was the first to represent on a diagram of energy *internal work*, and it is the purpose of the present paper to extend these representations to some cases not before given. In order to make the discussion complete it will be necessary to repeat some principles which may be more or less familiar to the reader. The representations are founded upon the following theorems:

*The mechanical equivalent of the heat absorbed or given out by a substance in passing from one given state (as to pressure and volume) to another given state, through a series of states represented by the coördinates of a given curve on a diagram of energy, is represented by the area included between the given curve and the curve of no transmission of heat drawn from its extremities, and indefinitely prolonged in the direction representing increase of volume.**

If a fluid be worked through a series of changes (as to pressure and volume) from any given state back to the same state, the resultant internal work will be zero.†

* Rankine, *Phil. Trans.*, 1854. *Steam Engine*, p. 303.

† Clausius on *Heat*.

Let AB (Fig. 202) be the path of the fluid, $A\varphi_1$ and $B\varphi_2$ adiabatics indefinitely extended to the right, H the heat absorbed in working from A to B; then, according to Rankine's theorem,

$$H = \varphi_1 AB\varphi_2.$$

If the fluid be imperfect, internal work will be done during the indefinite expansion $A\varphi_1$ which represent by the area $\varphi_1 Aan_1$. The more the gas is expanded the more it approximates to a

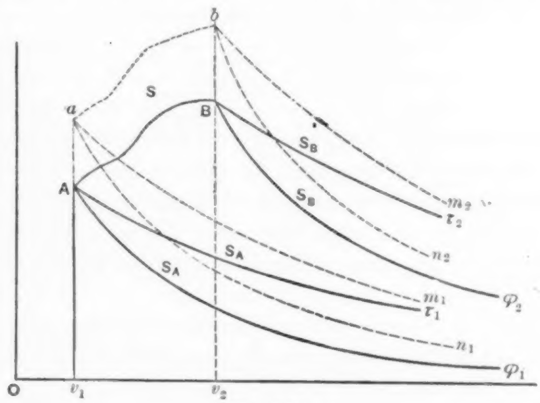

FIG. 202.

perfect gas and the internal work diminishes indefinitely, so that an_1 is asymptotic to $A\varphi_1$. Let ${}_AS_\infty$ represent this internal work; then

$${}_AS_\infty = \varphi_1 Aan_1.$$

Similarly,

$${}_BS_\infty = \varphi_2 Bbn_2.$$

The *external* work in passing from state A to state B will be:

$$U = v_1 ABv_2,$$

during which a certain amount of internal work will be done, which represent by

$${}_AS_B = ABba,$$

read "S between A and B," or "internal work between A and B."

Conceiving that φ_1 and φ_2 are infinitely distant, $A\varphi_1$ and $B\varphi_2$ asymptotic, the cycle will be closed, and will be $\varphi_1 AB\varphi_2$, while the internal work will be:

$$- {}_AS_\infty + {}_AS_B + {}_BS_\infty = \varphi_1 AB\varphi_2 n_2 ban_1 = o.$$

This is zero, according to the second theorem above. From this equation we have:

$$_AS_B = {_AS_\infty} - {_BS_\infty}.$$

The determination of $_AS_\infty$ and $_BS_\infty$ by means of adiabatic expansion is impossible in many, if not in all cases, and may be simplified for all cases by isothermal expansion as follows:

Through A and B pass the isothermals $A\tau_1$ and $B\tau_2$, and let $\tau_1 Aam_1$, indefinitely extended to the right, represent the internal work during the isothermal expansion, and $\tau_2 Bbm_2$ that during the isothermal expansion from B. Since the lines $A\tau_1$, am_1, etc., are asymptotic to each other, we have the closed cycles $\tau_1 AB\tau_2$ and m_1abm_2, while $_AS_B$ is the same as before; hence

$$_AS_B = \tau_1 Aam_1 - \tau_2 Bbm_2;$$

also,

$$Aan_1\varphi_1 A = Aam_1\tau_1 A = {_AS_\infty}:$$

that is,

$$_AS_\infty = \tau_1 Aam_1,$$

and

$$_BS_\infty = \tau_2 Bbm_2.$$

The heat absorbed may do three things:

1st. It may do external work.

2d. It may do internal work.

3d. It may change the energy of the substance by changing its temperature.

The general differential equations of thermo-dynamics are (the author's work, p. 48):

$$\left.\begin{aligned} dH &= K_v d\tau + \tau\left(\frac{dp}{d\tau}\right)_v dv, \\ dH &= K_p d\tau - \tau\left(\frac{dv}{d\tau}\right)_p dp, \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (A)$$

in which H is the heat absorbed, K_v the specific heat of the fluid at constant volume, τ the absolute temperature, v the volume of unity of weight, and p the pressure on unity of area. To transform the first of these equations so as to represent the three respective works mentioned above, add and subtract pdv, the external work, giving

$$H = \int pdv + \int\left[\tau\left(\frac{dp}{d\tau}\right)_v - p\right]dv + \int K_v d\tau, \quad . \quad . \quad . \quad (1)$$

the second term of the second member representing *internal* work; but it cannot generally be computed unless τ be constant, since otherwise the quantity in [ ] will not be a function of v only. The fluid may, however, simply *for the sake of determining the internal work*, be worked from A (Fig. 203) along the isothermal τ_1 to C and thence to B, in which case the [ ] becomes, by reduction, a function of v, since the temperature from A to C will be constant and may be represented by τ_1; and the preceding equation becomes

$$H = U + \int_{v_1}^{v_2}\left[\tau_1\left(\frac{dp}{d\tau}\right)_v - p\right]dv + \int_{\tau_1}^{\tau_2} K_v d\tau. \quad . \quad . \quad (2)$$

Represent the second term of second member by $ACcaA$. The third term cannot be integrated unless K_v be a known function of v; it generally involves internal work due to increase of tem-

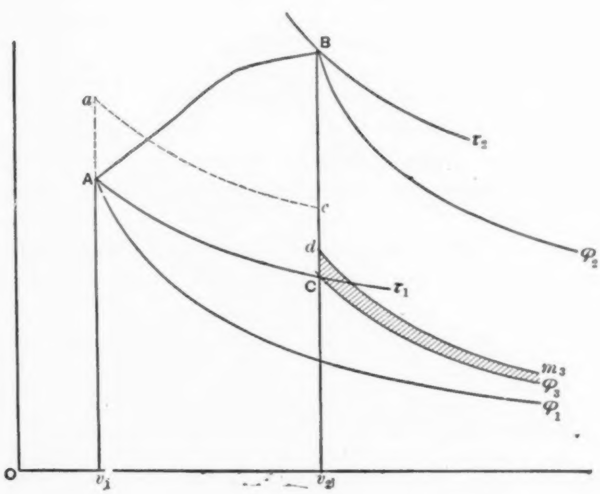

FIG. 203.

perature, and if this work could be eliminated the remainder of the heat required to change the temperature one degree would, according to an hypothesis of Rankine, be independent of the temperature or pressure, and hence would be constant. Conceive an isothermal drawn from any point between C and B

extending indefinitely to the right; then will the *internal* work due to an expansion along this isothermal be

$$S = \int_{\infty}^{v_2} \left[\tau \left(\frac{dp}{d\tau} \right) - p \right] dv. \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The *internal* work *due to a change* $d\tau$ *of the temperature* at the volume v_2 will be the differential of this expression, or

$$dS = \int_{\infty}^{v_2} \left[\tau \frac{d^2p}{d\tau^2} d\tau - d\tau \frac{dp}{d\tau} - dp \right] dv = \int_{\infty}^{v_2} \tau \frac{d^2p}{d\tau^2} d\tau dv, \quad . \quad (4)$$

and *the entire heat absorbed* for a change of temperature from τ_1 to τ_2 will be

$$\int_{\tau_1}^{\tau_2} \int_{\infty}^{v_2} \tau \frac{d^2p}{d\tau^2} d\tau dv + C_v \int_{\tau_1}^{\tau_2} d\tau = \int_{\tau_1}^{\tau_2} K_v d\tau. \quad . \quad . \quad . \quad (5)$$

This expression is represented by the area $\varphi_3 C B \varphi_2$ (Fig. 203), or $\varphi_1 A B \varphi_2$ (Fig. 204), the shaded part representing the first term

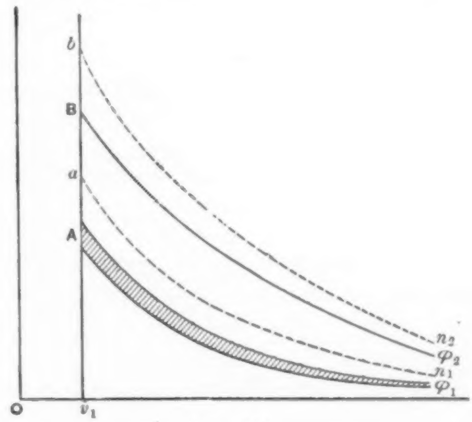

FIG. 204.

of the expression, or internal work, and the unshaded part representing $C_v(\tau_2 - \tau_1)$. But the shaded part is the difference of the internal works in passing out along isothermals through states A and B respectively and indefinitely, as in Fig. 205; or,

$$\int_{\tau_1}^{\tau_2} \int_{\infty}^{v_2} \tau \frac{d^2p}{d\tau^2} d\tau dv = \tau_1 Aam_1 - \tau_2 Bbm_2 = \textit{the shaded part},$$

all indefinitely extended. The unshaded part between $\tau_1 AB\tau_2$ in Fig. 205 is not the same as the unshaded part between $\varphi_1 AB\varphi_2$ in Fig. 204; but the shaded parts are equivalent.

Equation (2) becomes

$$H = U + \int_{v_1}^{v_2}\left[\tau_1\left(\frac{dp}{d\tau}\right) - p\right]dv + \int_{\tau_1}^{\tau_2}\int_{\infty}^{v_2}\tau\frac{d^2p}{d\tau^2}d\tau dv + C_v\left(\tau_2 - \tau_1\right) \tag{6}$$

which is separated into the three parts above enumerated.

The third term of the right member of this equation may be put in another form. The internal work in passing from C to

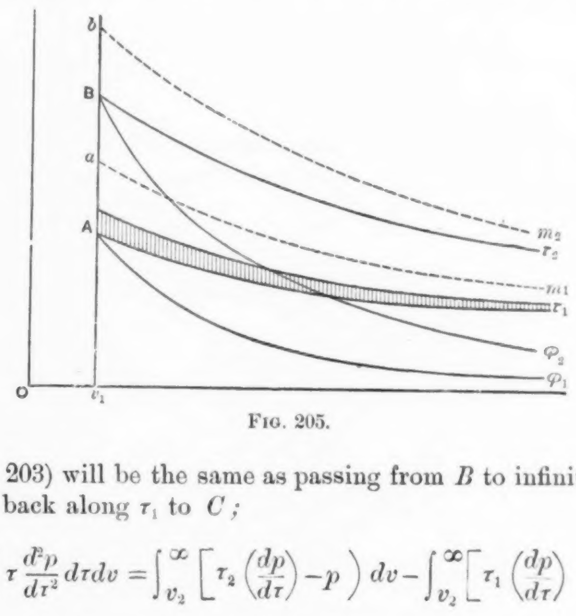

Fig. 205.

B (Fig. 203) will be the same as passing from B to infinity along τ_2 and back along τ_1 to C;

$$\therefore \iint \tau\frac{d^2p}{d\tau^2}d\tau dv = \int_{v_2}^{\infty}\left[\tau_2\left(\frac{dp}{d\tau}\right) - p\right)dv - \int_{v_2}^{\infty}\left[\tau_1\left(\frac{dp}{d\tau}\right) - p\right]dv$$

$$= \int_{\infty}^{v_2}\left[\tau_1\left(\frac{dp}{d\tau}\right) - p\right]dv - \int_{\infty}^{v_2}\left[\tau_2\left(\frac{dp}{d\tau}\right) - p\right]dv,$$

and this substituted in equation (6) gives, observing the proper limits,

$$H = U + \int_{\infty}^{v_1}\left[\tau_1\left(\frac{dp}{d\tau}\right) - p\right]dv - \int_{\infty}^{v_2}\left[\tau_2\left(\frac{dp}{d\tau}\right) - p\right]dv + C_v(\tau_2 - \tau_1) \tag{7}$$

$$= U + {}_AS_\infty - {}_BS_\infty + C_v(\tau_2 - \tau_1),$$

$$= V_1ABv_2 + {}_AS_B + C_v(\tau_2 - \tau_1), \text{ in Fig. 202.}$$
$$= V_1ABv_2 + AacC + \varphi_3CB\varphi_2, \text{ in Fig. 203.}$$
$$= V_1ABv_2 + AacC + \varphi_3\ Cdm_3 + C_v\ (\tau_2 - \tau_1), \text{ in Fig. 203.}$$

When the expansion is at constant temperature the case is comparatively simple. If the fluid be saturated vapor the isothermal will be a straight line parallel to the axis of v. In Fig. 206 let Ov_1 be the volume of a pound of water, and let it be evaporated at the constant pressure v_1A to the volume Ov_2, $A\varphi_1$ and $B\varphi_2$ adiabatics. The internal work of producing steam from water at three atmospheres is about eleven times the external work, and at one atmosphere it is some twelve times; hence at three atmos-

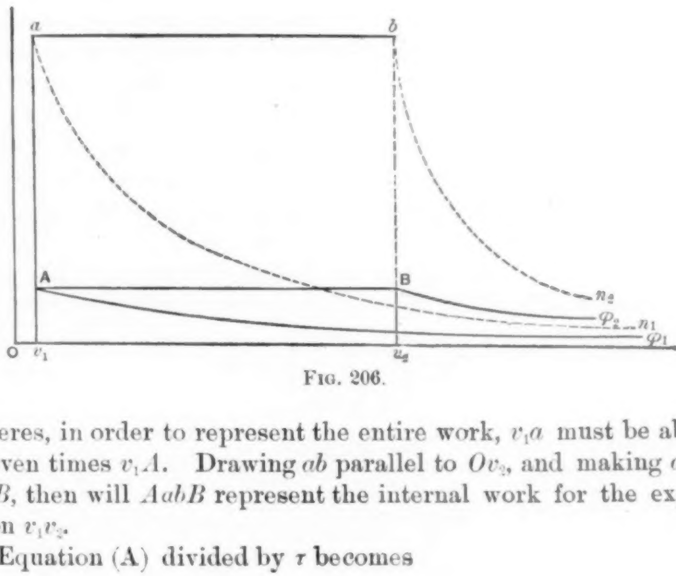

FIG. 206.

pheres, in order to represent the entire work, v_1a must be about eleven times v_1A. Drawing ab parallel to Ov_2, and making $ab = AB$, then will $AabB$ represent the internal work for the expansion v_1v_2.

Equation (A) divided by τ becomes

$$\frac{dH}{\tau} = K_v\frac{d\tau}{\tau} + \left(\frac{dp}{d\tau}\right)_v dv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (8)$$

Rankine calls the expression $\int\frac{dH}{\tau}$ *the thermo-dynamic function*, and represents it by φ; Clausius calls it *entropy*.

There is no good reason for the term used by Rankine, since any other function involved in the transferrence of heat might have received the same name with as much propriety, while the

term used by Clausius is generically appropriate. The former, however, is understood by all students of the science to apply to the same thing as the latter. Rankine does not define the term except algebraically; he simply writes:

$$\varphi = C_v \log \tau + \frac{dU}{d\tau}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (9)$$

where C_v is that part of the specific heat which goes to make the substance hotter, and $\frac{dU}{d\tau}$ the rate of doing work both external and internal per unit of temperature, due to an increase of temperature $d\tau$, for an expansion dv, and U the external work, and calls the expression the "thermo-dynamic function."

It may be defined as the sum of the quotients arising from dividing the elementary portions of heat absorbed by a body by the respective absolute temperatures at which it is absorbed.

The definition and conception are much simplified by confining it to an elementary quantity of heat, for then it may be defined as one of the equal parts resulting from dividing an elementary portion of heat by the absolute temperature at which it is absorbed. This will be represented by $d\varphi$; its value is given by equation (8).

To represent $d\varphi$ on a diagram of energy, let the path be arbitrary, as LM (Fig. 207). Through A pass an isothermal $A\tau$, and

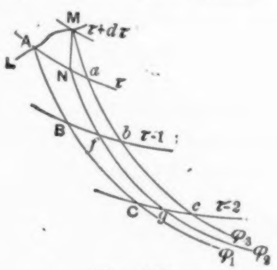

FIG. 207.

other isothermals below τ differing by one degree, marked $\tau-1$, $\tau-2$. Draw two consecutive adiabatics $\varphi_1 A$, $\varphi_3 M$. From M drop the perpendicular MN to the intersection N on the τ-isothermal, and draw the adiabatic $N\varphi_2$. Then will

$$\begin{aligned} dH &= \varphi_1 A N \varphi_2 + \varphi_2 N M \varphi_3 \\ &= \tau\left(\frac{dp}{d\tau}\right)_v dv + K_v d\tau. \end{aligned}$$

The isothermals differing by unity, divide each of the areas $\varphi_1 AN\varphi_2$ and $\varphi_2 NM\varphi_3$ below the τ-isothermal into equal parts, and the area AaM disappears, since it will be an infinitesimal of a lower order than $AabB$; hence

$$
\begin{aligned}
d\varphi = \frac{dH}{\tau} &= \left(\frac{dp}{d\tau}\right)_v dv + K_v \frac{d\tau}{\tau} \\
&= ANfB + Nabf \\
&= AabB = BbcC = \text{etc.}
\end{aligned}
$$

The *thermo-dynamic function* then is represented on a diagram of energy by the area between two consecutive adiabatics and two isothermals differing by one degree.

Since the heat absorbed in passing from A to M is represented by $\varphi_1 AM\varphi_3$, we may define :

The *thermo-dynamic function*, or $d\varphi$, is the rate of transferrence of heat per degree of absolute temperature, as a body absorbs or emits heat according to any law.

The equation of the gas—or fluid—must be known in order to determine the general value of entropy, and even if this be given the determination frequently involves complex analysis. We may illustrate the method by taking a perfect gas. The equation of the gas will be :

$$pv = R\tau;$$

$$\therefore \left(\frac{dp}{d\tau}\right)_v = \frac{R}{v}.$$

The entropy will be, the specific heat being constant,

$$\int d\varphi = C_v \int \frac{d\tau}{\tau} + \int R \frac{dv}{v}$$

$$\varphi_B - \varphi_A = C_v \log \frac{\tau_2}{\tau_1} + R \log \frac{v_2}{v_1}. \quad . \quad . \quad . \quad (10)$$

The most natural unit from which to reckon entropy would, at first thought, seem to be the absolute zero, but an inspection of the preceding equation shows that it would be reduced to infinity, for $\tau_1 = o$, and hence that limit is impractical; and since it is only the difference of the functions that is needed, no inferior limit is assigned. I designate this difference by the nota-

tion ${}_A\varphi_B$, read "φ between A and B," or "the entropy between A and B."

In Fig. 208 let AB be the "path of the fluid;" it is required to find the entropy between A and B. According to equation (10)

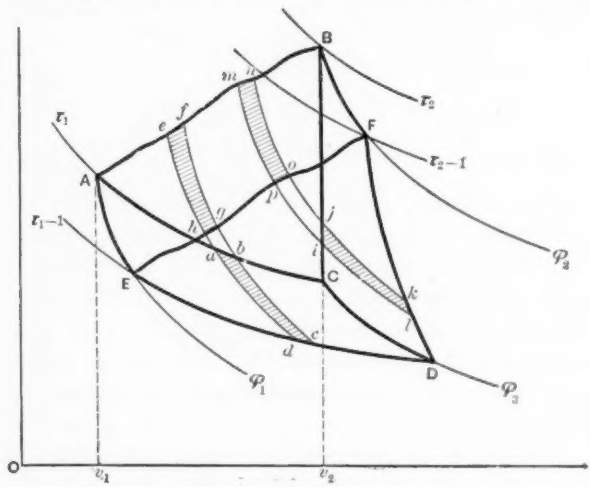

FIG. 208.

it may be separated into two parts, first with the volume constant and the temperature varying, and in the second the temperature constant and the volume varying. The latter will be the heat absorbed along an isothermal per unit of absolute temperature, from v_1 to v_2. Let AC be the τ_1-isothermal, ED an isothermal one degree lower, AE, CD and BF adiabatics, then will

$$ACDE = R \log \frac{v_2}{v_1} = {}_A\varphi_C$$

Let ml be any adiabatic passing between C and B, then will the heat absorbed for an increase of temperature $d\tau$ be the area between the consecutive adiabatics il and jk extended indefinitely to the right, or

$$C_v d\tau = lijk \text{ indefinitely extended.}$$

Cut the adiabatic il by an isothermal one degree lower than that at i, then will

$$\text{area } ilkj = C_v \frac{d\tau}{\tau}$$

$$\therefore CBFD = C_v \int^{\tau_2} \frac{d\tau}{\tau} = C_v \log \frac{\tau_2}{\tau_1} = {}_C\varphi_B.$$

Extend the adiabatics *da* and *cb* back to an intersection with the path *AB*, and cut *ed* by an isothermal one degree lower than that at *e* and let *h* be the point of intersection; then, by definition, *efgh* will equal *abcd*. Similarly, *mnop* will equal *ijkl*. By dividing the areas *ACDE* and *CBFD* into an indefinite number of strips by adiabatics, and finding the equivalent of the areas having their bases on the line *AB*, we will find an area *ABFE* equal to both the former areas. Hence, we have

$$_A\varphi_B = {_A\varphi_C} + {_C\varphi_B},$$

or,

$$ABFE = ACDE + CBFD$$

$$= R \log \frac{v_2}{v_1} + C_v \log \frac{\tau_2}{\tau_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

If the temperature be constant during expansion, several simple relations exist in terms of the thermo-dynamic function. Thus, if τ be constant the first of equations (A) becomes

$$d\varphi = \left(\frac{dp}{d\tau}\right)_v dv;$$

$$\therefore \left(\frac{d\varphi}{dv}\right)_\tau = \left(\frac{dp}{d\tau}\right)_v . 1 \quad . \quad . \quad . \quad . \quad . \quad (12)$$

In Fig. 209 let *ag* be an adiabatic and *am* an isothermal. They

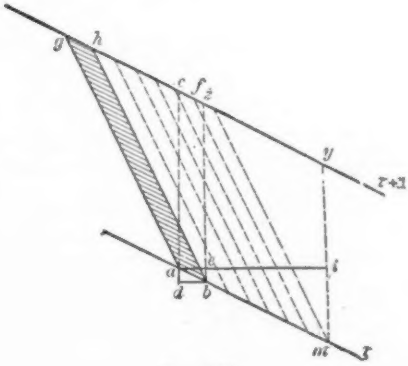

FIG. 209.

are here represented by right lines, since these relations exist only for a *state*, as *a*, and the ratios are *rates*. The points *a* and *b*

being consecutive, and gz an isothermal unity above am, the area $aghb$ will be $d\varphi$. $d\varphi \div dv$ will be the number of $d\varphi$'s in the abscissa unity. Pass along the isothermal am until the abscissa of m is $ai = 1$ (the unit being chosen arbitrarily); through m draw the adiabatic mz; then will the area $gamz$ contain the $d\varphi$'s for $v = 1$. In the right member $dp \div d\tau$ for v constant is the vertical distance between two isothermals differing by unity, and hence is ac; and this multiplied by $v = 1$ gives the area of the trapezoid $acym$. But this trapezoid has the same base am as $amzg$, and between the same parallels; hence they are equal.

It will be observed that in equation (12) the subscript τ of the left member is the variable in the denominator of the right member, and vice-versa. Hence taking the reciprocals of equation (12) we have,

$$1.\left(\frac{dv}{d\varphi}\right)_p = \left(\frac{d\tau}{dp}\right)_\varphi .1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

In Fig. 210 let $agzk$ be the arbitrary area representing ${}_a\varphi_k = 1$; then will the abscissa of k from a with p constant be ai. If $ao = p = 1$, then will $aovi$ represent the left member of the equation;

$$\therefore 1.\left(\frac{dv}{d\varphi}\right)_p = aovi = aouk = atnk.$$

In the right member $\left(\frac{d\tau}{dp}\right)_\varphi$ is the increase of temperature for an increase of $p = 1$; hence it gives the increase of tempera-

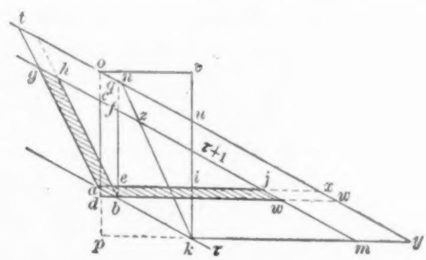

FIG. 210.

ture of the isothermal tn above that of ak, and since $agzk$ is unity and is between two isothermals differing by unity, multiply-

ing it by $\left(\frac{d\tau}{dp}\right)_\varphi$ gives the area *atnk*, the same as just found for the left member.

The second of equations (A) for τ constant becomes:

$$d\varphi_\tau = -\left(\frac{dv}{d\tau}\right)_p dp;$$

$$\therefore \left(\frac{d\varphi}{dp}\right)_\tau = -\left(\frac{dv}{d\tau}\right)_p .1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The left member is the $d\varphi$'s for p unity. Pass along the isothermal *ak* until the ordinate of k in reference to a is unity; it will be negative. Draw *aj* horizontal, intersecting the $\tau + 1$ isothermal at j, then

$$d\varphi = aghb = ajwb,$$

and drawing *zk* parallel to *ag* and *km* parallel to *aj*; then $agzk = ajmk$, which is the value of the left member. In the right member $\left(\frac{dv}{d\tau}\right)_p$ is the value of v for one degree of temperature, and hence is *aj*; and this multiplied by $p = 1 = ap$ gives *ajmk*, the same value before found.

Taking the reciprocal of equation (14) we have

$$1.\left(\frac{dp}{d\varphi}\right)_v = -\left(\frac{d\tau}{dv}\right)_\phi .1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (15.)$$

In Fig. 210 let $agzk = ajmk = \varphi = 1$, and $ax = v = 1$. Through x draw the isothermal *txy*. In the left member of equation (15) $dp \div d\varphi$ will be *ap*, and is negative, and this multiplied by $ax = v = 1$ gives the area of the trapezoid $axyk = atnk$.

The right member is the difference in temperature of the isothermals *ak* and *tx*, which, multiplied by $\varphi = 1 = gzka$, gives the area *tnka* as before.

Had the origin been at k then would the ordinate of a, or *pa*, been positive, but v would have been negative, so that the negative sign in equations (14) and (15) would remain.

If τ be constant the first of equations (A) gives

$$\left(\frac{dH}{dv}\right)_\tau = \tau\left(\frac{dp}{d\tau}\right)_v .1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

The left member of this equation is the heat absorbed at constant temperature while expanding unity of volume, provided the amount so absorbed is the same for each elementary increase of volume. $dH \div dv$ being a *rate*, is shown as constant by making the path of the fluid a straight line. In Fig. 211 let

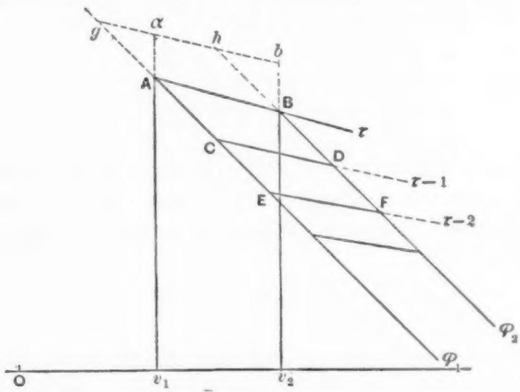

FIG. 211.

AB be the path of the fluid, limited by the ordinates v_1A and v_2B where $v_2 - v_1$ is unity. $A\varphi_1$, $B\varphi_2$ being adiabatics, $\varphi_1 AB\varphi_2$ will be the heat absorbed, and is $dH \div dv$. In the right member $\left(\frac{dp}{d\tau}\right)_v$ is the increase of pressure for an increase of one degree of temperature. Let gb be an isothermal one degree higher than AB; then will the ordinate

$$Aa = \left(\frac{dp}{d\tau}\right)_v,$$

which being multiplied by $v = 1 = v_2 - v_1$ gives the area $AabB$; hence,

$$\left(\frac{dp}{d\tau}\right)_v (v = 1) = AabB = AghB = ABCD, \text{ etc.}$$

$$\therefore \tau \left(\frac{dp}{d\tau}\right)_v .1 = \varphi_1 AB\varphi_2 .$$

Differentiating equation (16) regarding τ as variable gives

$$\frac{d}{d\tau}\left(\frac{dH}{dv}\right)_\tau = \tau\left(\frac{d^2p}{d\tau^2}\right)_v .1_v + \left(\frac{dp}{d\tau}\right)_v .1_v ; \quad . \quad . \quad . \quad (17)$$

where the subscript v in the factor 1_v indicates the quantity whose value is unity. The left member of (17) is the increased amount of heat which must be absorbed by increasing the temperature one degree, the expansion being for an increase of unity of volume at the higher temperature. In Fig. 212 if AB

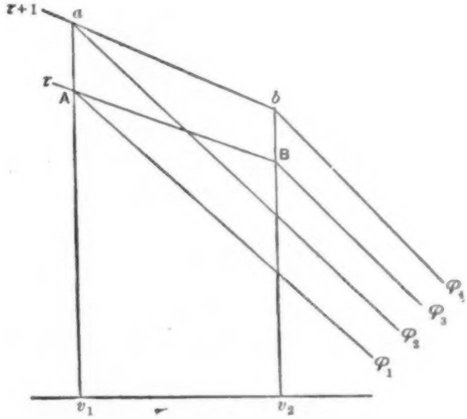

FIG. 212.

be a τ isothermal, and $v_2 - v_1 = 1$, then, as shown in Fig. 211, will

$$\left(\frac{dH}{dv}\right)_\tau = \varphi_1 AB\varphi_3.$$

Let ab be the $\tau + 1$ isothermal, then

$$\frac{d}{d\tau}\left(\frac{dH}{dv}\right)_\tau$$

$$= \varphi_1 Aa\varphi_2 + \varphi_2 ab\varphi_4 - \varphi_1 AB\varphi_3 - \varphi_3 Bb\varphi_4.$$

$$= (\varphi_1 Aa\varphi_2 - \varphi_3 Bb\varphi_4) + (\varphi_2 ab\varphi_4 - \varphi_1 AB\varphi_3).$$

$$= \tau\left(\frac{d^2p}{d\tau^2}\right)_v .1_v + \left(\frac{dp}{d\tau}\right)_v .1_v,$$

which was to be proved.

The second of the equations (A) for τ constant gives

$$\left(\frac{dH}{dp}\right)_\tau = -\tau\left(\frac{dv}{d\tau}\right)_p .1_p \quad . \quad . \quad . \quad . \qquad (18)$$

The left member implies that one is to pass along the τ isothermal until there is a diminution of pressure of unity.

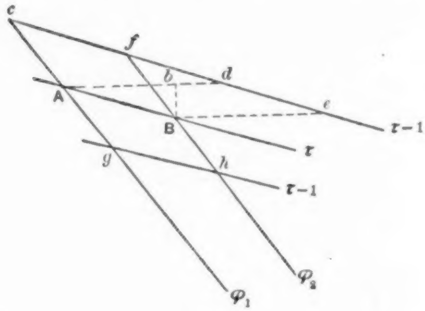

FIG. 213.

In Fig. 213 let AB be the τ isothermal; Ad, Be, horizontals; cf the $\tau + 1$ isothermal. The trapezoids $AdeB$ and $ABfc$ are equal, and

$$\left(\frac{dH}{dp}\right)_\tau = \varphi_1 AB \varphi_2.$$

We also have $$\left(\frac{dv}{d\tau}\right)_p = Ad;$$

$$\therefore \left(\frac{dv}{d\tau}\right)_p .1_p = AdeB = AcfB;$$

$$\therefore \tau \left(\frac{dv}{d\tau}\right)_p .1_p = \varphi_1 AB \varphi_2,$$

as before.

CCCCIV.

ON THE MEASUREMENT OF DURABILITY OF LUBRICANTS.

BY J. E. DENTON, HOBOKEN, N. J.
(Member of the Society.)

PRACTICAL differences of durability of lubricants depend not on any differences of inherent ability to resist being "worn out" by rubbing, but upon the rate at which they flow through and away from the bearing surfaces. The conditions which control this flow are so delicate in their influence that all attempts thus far made to measure durability of lubricants may be said to have failed to make distinctions of lubricating value having any practical significance. To illustrate: let a standard half-inch plug-gauge and its ring be thoroughly cleansed of all oily matter by soaking in ether. If the two pieces are fitted within the limit of standard accuracy there will be, say, about one thirty-thousandth of an inch greater diameter to the ring than to the plug. The two pieces will go together with great difficulty after cleansing with ether—in general a mallet will need to be used to force the plug into the ring. But if the pieces be smeared with lard or sperm oil, or any good oil of equal fluidity, the plug may be pushed within the ring by a slight effort of the hand, and may be moved back and forth at the rate of about twenty times per minute with the utmost effort of the wrist. If the reciprocating motion is stopped for five seconds, the plug will stick and must be driven out of the ring with a mallet. The phenomena to be here noticed are, first, that the oil allows itself to be swept into the crack between the ring and the plug by the reciprocating motion of the latter, thereby acting like a wedge to expand the ring slightly; and, second, that as soon as the motion of the plug ceases the oil between the ring and the plug is squeezed out by the elastic recovery of the ring. The orifice through which the oil flows from underneath the ring is a circular annulus one sixty-thousandth of an

inch wide, and yet the flow is a perfectly definite one, inasmuch as the phenomena described repeat themselves with fair regularity.

Let both pieces be again cleansed in ether and the experiment repeated, using as the lubricant an oil so thick as scarcely to flow out of any vessel at ordinary temperature. Such an oil is found in the petroleum products prepared for steam-cylinder lubrication. It will be found that the plug will enter the ring with the same readiness as when lard was the lubricant, but the force of the wrist will not be able to reciprocate the plug more than about one-third as fast, owing to the greater viscosity of the heavy oil. If the reciprocating movement of the plug is stopped, a period of at least fifteen seconds is necessary to cause the plug to stick, so that the hand cannot readily restart it. The explanation of this difference of action of the two lubricants is simply that more time is required for the heavy oil to escape underneath the ring and allow metallic contact between the latter and the plug to be partially resumed, and that the thicker oil requires more force to overcome the friction among its own particles. The difference is proportional to the difference of viscosity as determined by the time required for each to flow through any orifice, and to their coefficient of friction as determined by any form of oil-testing apparatus capable of measuring friction with a superabundance of oil fed to the rubbing surfaces. The sticking of the plug, when it is allowed to rest, has also its parallel in the excessive friction offered by a journal to starting compared to its friction when in uniform motion, the oil squeezing out from between a journal and its brass when the former comes to rest in the same manner that it flows from underneath the ring-gauge. It is evident, however, that if these definite differences of flow of a lubricant from between two surfaces exist for such minute thicknesses of film as occur with the plug and the ring and for such slow motion as the hand produces, in studying the action of lubricants in practice we must adjust our observations and scale of measurement to the consideration of the most delicate and subtle forces or influences to which the lubricant can possibly be subjected in its use in practical service.

From this standpoint let us consider the results of some attempts to measure the durability of oil in the laboratory. The experiences given are those of the writer, and they are set forth in detail to an extent designed to contribute in what, it is hoped,

will be a useful way to the inquiries and beliefs which are current regarding possibilities in the way of practical distinctions of lubricating value by such forms of testing machines as have thus far been used in testing laboratories. The primary idea of determining durability is, naturally, to supply a measured quantity of lubricant to certain rubbing surfaces and determine how great an amount of rubbing the lubricant will withstand before its power is exhausted of maintaining the friction at some minimum agreed upon. As a means of measuring the friction between two rubbing surfaces there is no device superior to what is known as the Thurston form of oil-tester, which consists in forcing a pair of brasses against a journal in opposite directions by a spring, the latter being lodged in a pendulum free to swing about the journal under the influence of the friction produced between the brasses and the latter, the amount of the friction being measured by the inclination to the vertical of a line joining the centre of the journal and the centre of gravity of the pendulum. Suppose one of these oil-testing machines to have a journal about $1\frac{1}{2}$ inches square; then an amount of oil forming a single drop let fall from the sharp point of a steel rod, $\frac{1}{16}$ of an inch in diameter, suffices to cover the rubbing surfaces, so that with the latter in the best possible condition a very low coefficient of friction can be obtained at a rubbing speed of about 200 feet per minute and a pressure of 75 lbs. per square inch of the journal; and at the same time there will be no surplus oil—that is, all of the oil will be practically subjected to rubbing action. If, under these circumstances, we run the journal continuously until the friction, after remaining practically constant for considerable time—half an hour at least—gradually begins to increase and finally becomes double the original amount, it might easily be supposed that the number of revolutions to accomplish this would measure the durability of the lubricant used as compared to another tested in the same manner. This was, at one time, so far the belief of the writer that the relative value of oils was sought by such a test, but with the following result: The amount of oil applied to the journal was accurately gauged at 8 milligrammes. The journal and brasses—the former of tool steel, hardened, and the latter of composition—were reduced to such a condition that at a given temperature a certain standard lard oil always produced a certain amount of friction, and a trial of this standard lard always preceded the trial of any other oils whose durability was to be determined

relatively to the lard.* If the friction with the standard lard was too great, the cause of it was always found ascribable to incipient roughening of the surfaces, and the latter were nursed by rubbing with an oil-stone under a magnifying glass and by continued running with an unlimited amount of lubricant, until the standard condition was regained. When the oil was fresh upon the bearing surfaces it was translucent in appearance and slippery to the touch. After about 10,000 revolutions the friction might have increased to double the minimum amount, as described above, but the appearance of the oil would show no signs of deterioration, and it was found that the cause of the extra friction was the uneven distribution of the oil over the rubbing surfaces. There being no continuous supply, the oil distributed itself more or less evenly simply by chance, and a small filament or part of the rubbing surfaces could be seen to be devoid of oil. Accordingly the oil was redistributed with the pointed end of a match and the pressure reapplied, with the result that the friction at once resumed the minimum value, showing that the oil was in no sense worn out. The machine would therefore be run on until the friction again doubled itself, when another examination might still show the cause to be irregular distribution. This was again remedied and the machine continued in operation under the same programme, until finally, after several redistributions of the oil,

* A question often asked is whether measurements on an oil-testing machine can be made to duplicate themselves. In the matter of *friction* an entirely satisfactory duplication of action is always possible. For example, the following is a series of fifteen consecutive tests of different oils, and between each pair of tests the journal of the oil-tester was tried with the standard oil, and gave results as follows:

No. of Trial.		Temperature of Bearings.	Friction of Standard Oil.	
First Trial, December 15th	Oil A.	95°	1.9 lbs.	Total pressure on journal, 300 lbs.
Second " " 18th		95°	1.9 "	
Third " " 25th		110°	1.7 "	
First " " 15th	" B.	95°	1.9 "	
Second " " 27th		110°	1.7 "	
First " " 16th	" C.	95°	1.9 "	
Second " " 22d		95°	1.9 "	
First " " 15th	" D.	95°	1.9 "	
Second " " 18th		110°	1.7 "	
First " " 18th	" E.	110°	1.7 "	
Second " " 23d		110°	1.8 "	
First " " 18th	" F.	110°	1.7 "	
Second " " 18th		110°	1.7 "	
First " " 18th	" G.	110°	1.7 "	
Second " " 28th		110°	1.7 "	

the latter had lost its translucent appearance and oily nature, having become a dark-colored paste. The friction would then refuse to decrease by redistribution. The total number of revolutions might be 12,000 to accomplish this result with lard oil on a given date, and the same number of revolutions might be nearly obtained on a second trial. In fact, so far did duplicate trials of the same oil apparently give the same revolutions within a small per cent., that, as stated above, the relative durability was thought to have been determined by this method with reference to lard oil. But, upon a more extensive trial of the lard, 30,000 revolutions were required to reduce it to the gummy state, and again, 60,000 revolutions, and upon one occasion 400,000 revolutions; and in between these figures would occur records of 10,000, 7,000, and as low as 5,000 turns of the machine, which, with no variation of pressure, temperature, speed, atmosphere, amount of oil or quality of the latter, were the apparent measure of the durability of this standard oil, as measured on this basis. Evidently some cause was at work which rendered the method valueless. Such cause was discovered and communicated to the writer by Dr. Charles B. Dudley, of Altoona, who had carried out similar investigations. Upon analyzing the black paste to which the lubricant reduced itself, Dr. Dudley found that it was composed mainly of the material of the bearings themselves, showing thereby that the limit to the friction-producing qualities of a lubricant tested in this manner was the rate at which it adulterated itself with the metal worn off the bearings, notwithstanding that the amount of the latter was so infinitesimal that years of operation of the machine would not detect the wear by any finite measurement of the dimensions of the bearings. The rate of this metallic wear being infinitely variable, it is evident that the durability of oil cannot be measured by restricting our measurement to the behavior of a fixed quantity whose amount shall be so small as to allow of no surplus oil about the bearings; or, in other words, maintain all of the oil under rubbing action at once. The failure of the method is attributable also to the fact that so little oil was employed compared to the amount of rubbing surface that the latter could not remain uniformly covered. It is natural, therefore, to inquire whether the method would not be more successful if a greater amount of oil could be used permitting a slight surplus, but applied upon some plan which would enable the surplus to be employed to secure regularity in the distribution of the

oil over the rubbing surfaces. Such a plan has been tried by the writer on a larger size of Thurston testing machine having a journal about four inches diameter and seven inches long, with brasses of the same proportions as used in railroad car service, except that the arc of contact was reduced so that the total bearing area of two brasses was about seventeen square inches. The journal was of wrought iron reduced to such a condition that with standard lard oil under 350 pounds per square inch pressure, and about 250 revolutions per minute, a coefficient of friction of four-tenths of one per cent. could be maintained. Oil was fed to the journal through two oil cups fastened into holes in the upper brass, as shown at *LL* in Fig. 214, which shows a longitudinal section of the testing journal. After cleaning the bearing surfaces, 10 minims of oil were smeared upon them, and the pendulum hung in place. Twenty-five minims of oil were then evenly distributed between the two oil cups, while about 100 lbs. pressure was upon the bearings and the machine in motion. The pressure was then gradually increased to 350 lbs., and the machine run until the minimum friction doubled itself. Before relating the results we will consider the difference in the amount of oil applied compared to that used in the eight-milligram test, with the journal about one and one-half inches diameter. For the latter conditions each 10,000 revolutions causes a number of square inches of surface to be rubbed over which correspond to the use of one milligram of oil for each 7,000 square inches. In the case of the larger journal 35 minims of oil correspond to the use of one milligram to each 2,000 square inches of surface rubbed over, if the 35 minims should permit 10,000 revolutions before the friction was doubled. Consequently, a considerable surplus of oil would be present in the case of the large

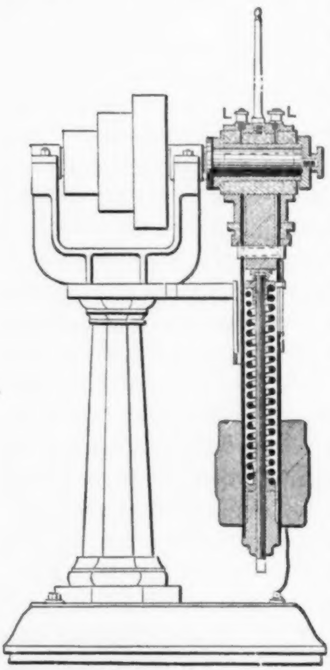

FIG. 214.

journal, provided it could be prevented from wasting. To prevent such waste was, however, a matter of some study. The 25 minims placed in the oil cups filled the lower portions of the latter for a depth of about one inch and flowed out very gradually, but still with sufficient abundance to cause large drops of oil to roll off of the rising side of the journal as they approached the edge of the upper brass. Oil adhering to this edge of the brass was carried by capillary action, and the slight irritation of the upper brass in its framework, up the vertical sides of the latter; and gradually the entire interior of the framework enclosing the brasses was covered with the surplus oil. To overcome this irregularity the upper brass was cut to the form shown in Fig. 215, so that oil which was carried upward by the journal to the edge of the brass, *a*, could not climb along the bevelled surface, *a b*, but dropped off of the upper brass to the lower brass. A flexible metal guide, *c d*, was fastened to the lower brass (Figs. 215 and 216), so that the oil thus dropping from the upper brass was prevented from striking the sides of the framework, but was instead caused to fall against the journal and be repeatedly drawn upward, thus assisting to maintain in a uniform condition the film of oil on the journal. The brasses were one inch shorter than the journal, so that a liberal amount of lateral motion * could be given to the pendulum. The edges of the brasses were chamfered, so that the result of the lateral movement was to accumulate a band of oil around each end of the journal distant from the latter equal to the radius of the fillet. This band of oil absorbed a considerable proportion of the total 35 minims during the early stages of

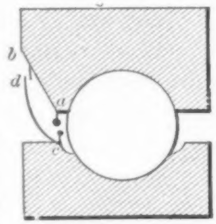

FIG. 215.

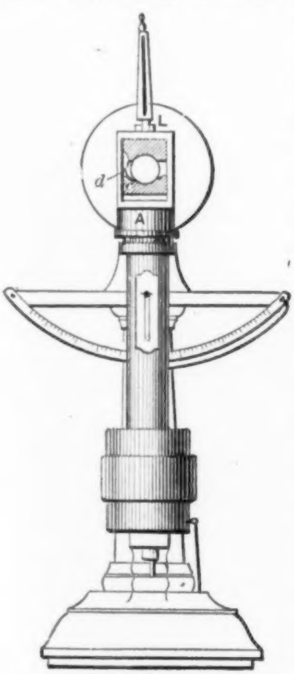

FIG. 216.

* Whenever this lateral motion was stopped the friction rapidly increased.

a test, but did not seem to consume oil or withdraw it from action between the surfaces after such time. To all appearances the journal was in a uniform condition of lubrication, the friction remaining perfectly constant for several hours, and the temperature remaining, with all oils of about the fluidity of lard, within 5° of 130° Fahr. Yet in spite of every precaution successive records of lard oil were as follows:

July 9	82,000 revs.
" 10	67,800 "
" 13	53,000 "
" 15	29,000 "

and of a pure mineral oil of about the same fluidity as lard:

July 10	36,000 revs.
Second test	33,000 "
Third "	27,000 "

The last three results were well calculated to lure an experimenter into the belief that the method gave duplicate results, but on continuing the test of the same oil on July 15, the record of one test was 10,600 revolutions, and of a second test 64,000 revolutions with the friction, and temperature, and speed as identical as it is possible to have them. It was evident, therefore, that irregularity of conditions existed of a nature beyond the reach of any available means of control, and so far as is known to the writer there is no method of employing so little oil on a journal as to prevent its escape from the latter and determining anything representative of the quality of durability. The clever device adopted in the Pennsylvania laboratory, at Altoona, of saturating a strip of felt laid in a cavity in the lower brass with the oil to be tested, is another method of making a fixed quantity of oil be wholly devoted to the lubrication of the bearing surfaces, the slight tendency of the oil to escape being checked by sucking up the waste oil with a pipette and continuously restoring it to the journal. No attempt is made, however, to determine any quality of the lubricants tested other than their coefficient of friction. The rates at which oils are supplied to practical bearings are, however, greatly in excess of the amount applied in the above tests. Taking the average revolutions which caused the friction to double at 30,000, the 35 minims supply to a journal four inches diameter and six inches long amounts to one seven-hundredth of a milligram per square inch of surface rubbed over. A very economical case of practical oil consumption is when a loco-

motive main crank-pin consumes about six cubic inches of oil in a thousand miles of service. This is equivalent to a consumption of one milligram to seventy square inches of surface rubbed over, or ten times as great an amount as that used on the testing machine. If, therefore, the latter amount provided a surplus requiring special provision to prevent escape of oil from the bearings, what must become of the surplus present on the locomotive crank-pin and other bearings using oil with the same economy, of which there are numerous examples in every class of machinery? Undoubtedly it must continuously escape from the journal. The 35 minims of oil, used in the test described, was so little contaminated by the absorption of metallic wear under 30,000 revolutions that the fluidity was not greatly reduced. Consequently, with ten times as great an amount of oil to absorb the metallic particles, the oil leaving the locomotive crank-pin is but little changed in fluidity, and flows from between the surfaces with practically the same freedom as when in a fresh condition. It follows, therefore, that the economy of one oil over another, so far as the quantity used is concerned—that is, so far as durability is concerned—is simply proportional to the rate at which it can insinuate itself into and flow out of minute orifices or cracks. Oils will differ in their ability to do this, first, in proportion to their viscosity, and, second, in proportion to the capillary properties which they may possess by virtue of the particular ingredients used in their composition. Where the thickness of film between rubbing surfaces must be so great that large amounts of oil pass through bearings in a given time, and the surroundings are such as to permit oil to be fed at high temperatures or applied by a method not requiring a perfect fluidity, it is probable that the least amount of oil will be used when the viscosity is as great as in the petroleum cylinder stocks. This was pointed out by the writer in a former paper presented to the Society in connection with a description of various viscosimeters. When, however, the oil must flow freely at ordinary temperatures and the feed of oil is restricted, as in the case of crank-pin bearings, it is not practicable to feed such heavy oils in a satisfactory manner. Oils of less viscosity or of a fluidity approximating to lard oil must then be used, and the differences among such oils, with respect to the quality of entering and leaving the spaces between the bearing surfaces, may readily be imagined to depend in a considerable degree upon differences of capillary property arising

from choice of ingredients forming any particular mixture. We are led, therefore, to conclude that a test of durability must be a test of the rate at which two oils fed to the same journal, by the same device, and under the same circumstances, permit themselves to flow between the bearing surfaces and out of, and off of, the bearings, and that oil may differ in this respect with every variation of ingredient and viscosity. As necessarily there will be surplus oil which will escape from the reservoir and do no lubricating service before being led off of the bearings, it follows that every particular condition of bearings will affect the amount of oil consumed, and hence our oil-testing machine, to determine durability, must be the very machine upon which the oil is used or a copy of the same.

To illustrate what would happen if this idea of testing durability was carried out with the large-sized Thurston testing machine referred to above, experiments were made feeding oil to the two oil cups in 50 minim doses under the same conditions as described above, except that the brasses were exactly like railroad car brasses, no provision being made to prevent the oil escaping in whatever direction it chose to flow. The machine was run until the friction began to increase above a minimum, and then another dose of 45 minims of oil was added to the cups without stopping or disturbing the machine. The oil was so regularly led away by the various escaping surfaces formed by the brasses, that in a series of about a dozen trials of lard oil versus sperm oil the latter averaged 11,000 revolutions in doubling the friction, and the lard 9,000 revolutions, with a variation of only about 20% from these figures in the various separate trials. But upon changing the brasses to another pendulum-head or framework, in which they fitted to all appearances with precisely the same degree of accuracy, a series of trials with lard oil averaged 14,000 revolutions to consume 50 minims. The slight difference in the fitting of the brasses in the pendulum-head by altering the effect of the various escaping surfaces increased the time necessary for the oil to escape. It seems hardly possible, therefore, that the quality of durability in lubricants can be measured, except by copying in a very perfect manner the exact conditions of arrangement, proportion and method of feeding which are to be used where the lubricant is to be applied; and even when this precaution is taken, the fact that in some kinds of service the limit to the consumption of oil depends upon the extent to which dust or other refuse becomes

mixed with it, as in railroad-car lubrication and in the case of agricultural machinery, makes it almost hopeless to expect that laboratory tests can contribute very much of practical value in the determination of relative durability of lubricants.

That machines may be devised for the laboratory, however, and are, in fact, in use to-day in some of the latter, which will contribute much valuable knowledge to many questions of lubrication, through measurements of friction and metallic wear, is not to be denied, but the discussion of this part of the subject is not intended to be included in the scope of this paper.

DISCUSSION.

Mr. T. R. Almond.—I would like to relate something which has come under my observation in connection with this subject. If the lubricant be enclosed in a case so that none of it can escape, and there be a perfect circulation of the lubricant running all over the bearings, and the lubricant be used for a lengthened period—one year, if you like—and the amount of work which is being done be considerable, the lubricant, instead of being inferior in quality as a lubricant at the end of that period of time, will prove to be superior in quality. An illustration of this came to my notice in a machine which I have had in use in my shop. We put a pint and a half of oil in the machine, and at the end of a year we took the oil out and replaced it with a pint and a half of the same oil as previously used. It came out of the same can. Upon starting the machine we found that it heated up, I should think from 10° to 15°, and continued to do so during a period of three or four weeks. After that it began to cool down to the same condition in which it was previous to our taking out the old oil, which had been used for twelve months. The kind of oil used was that called electric machine oil, made by the Vacuum Oil Company, of Rochester. The fact was well observed and beyond dispute.

Mr. H. A. Porterfield.—I cannot let this opportunity pass without saying a few words about the importance of two new views presented in this paper. Those two new views are the escape of lubricants and the effect of such escape on the endurance of them, and the effect on the endurance of a mixture of the lubricants with parts of the bearings themselves or with outside dust.

A series of tests of oils on a Thurston oil-testing machine at Cambria Iron Works, made to determine coefficients of friction

only, led me to think that the machine is valuable only for determinations of friction coefficients, and not for endurance tests.

The statements by the writer that endurance tests are affected by the escape of the lubricant from the bearing, and also by its mixing with particles of matter from the bearing itself or from outside sources, are particularly interesting to large plants, such as iron and steel works, where it is impossible to use the best grade of oil for each purpose, but where one or two oils only can be handled to answer many requirements.

In practice there are many bearings where an oil does not get a chance to show its enduring qualities, and on such bearings it is often economical to use a low-priced oil of less enduring qualities than the high-priced. This fact is worthy of consideration in the adoption by a large plant of an oil to be used for different purposes.

I recently saw an engine running in a place where there was a great deal of dust of the nature of iron. The oil used on the bearings was of high viscosity, and has the reputation of being one of the best and highest-priced mineral oils in the market. On substituting a second oil of one-third the viscosity and half the price of the first, an oil which, I think, if it could be got at properly, would have less enduring qualities than the first oil, the engine ran cooler on a little less quantity of the second oil than of the first. This may be due to the fact that the less-viscous oil mixed differently with the dirty matter and carried more ot it off the bearings.

Mr. Jno. T. Hawkins.—We have recently had examples of the use of lubricants under somewhat different conditions from those which obtained when the Thurston oil-testing machines were introduced and the tests made upon them. In such machines, for instance, as the Westinghouse engine and similar machines. I notice in the paper that under the Thurston tests, when the lubricant becomes discolored the friction becomes greater, and from that time on it possesses diminishing value as a lubricant.

In the Thurston machine the tests consist in some cases in what may be called a wearing out of a small, definite, measured quantity of the lubricant, the result being observed until it practically disappears.

In such machines as the Westinghouse engine, and indeed in the case of many other large journals, such as car-axle boxes and journals, the supply of lubricant is practically unlimited, and in the former case it is so confined to its reservoir that the whole bulk probably suffers some such deterioration as that indicated in

the paper as becoming discolored, but is not subject to the conditions imposed by the Thurston tests, which I have called "wearing out," or at least it cannot disappear, it would seem, under conditions exactly similar to its disappearance in the Thurston machine.

I would like to ask if any member has had experience in a case such as the Westinghouse engine which would enable us to determine whether or not the unlimited supply affects the question of deterioration in the lubricant and its disappearance; whether or not an equal quantity is worn out or consumed under such conditions of unlimited supply without escape, as would be in the case where only the quantity necessary to the best degree of lubrication is supplied to the bearings as fast as it disappears or is worn out, and whether the whole bulk of oil in the reservoir of a Westinghouse engine requires to be renewed at certain intervals, or merely added to from time to time.

Mr. F. A. Scheffler.—I can only say this much, that many of these engines of various sizes are run a long while without any additional oil being placed in the crank-case. The principal reason, I presume, why the oil has to be renewed is not that it loses its lubricating qualities, but simply because the oil gradually rises up and will overflow, owing to an increased quantity of water in the case from condensed steam, which leaks over sometimes into the crank-case. This action will raise the oil up after a long run, and the oil will have to be renewed in about six months or so.

Mr. C. S. Dutton.—The oil supply in the practical operation of the Westinghouse engine is kept up pretty regularly. A special oil made for that purpose, to stand a high degree of heat, is fed into the bearings—a sort of sight-feed arrangement usually—and it feeds itself into the inside of the crank-case. The constant agitation mixes it up very thoroughly with the water, and the constant leakage of condensed steam into the crank-case causes an overflow, and it takes the oil out with it to a certain extent, so that the supply has to be constantly kept up. But it is also true that in very many cases they catch the oil at the overflow and put it back and use it over to a considerable extent. I do not know so much about that. I have not been connected with Westinghouse engines for the last four or five years, but that is the practical operation of it. What the effect on the oil eventually is I could not say There is a certain amount of it which goes to waste, so that some additional oil has to be added all the time.

Mr. Porterfield.—I have seen a Westinghouse engine in which about 75% of the oil which escaped was allowed to settle out in the purifier, as it is called, and that oil did not seem to be changed to any extent from the original oil. It was perfectly good and could be put right back into the crank-case.

Mr. Almond.—Perhaps I have not given a fair description of the apparatus of which I was speaking. I wish to say that there are 11 bearings of steel and cast-iron in the machine; that I put in a pint and a half of vacuum electric machine oil; that it ran for 12 months at the rate of 140 revolutions a minute; that the machine was transmitting between 4 and 5 H.P. for 12 months; that at the end of that time one pint of oil was taken out, being all the oil which could be drained out of it. Therefore one-half pint was the amount consumed. The point to which I particularly wish to call the members' attention is that the machine ran warm after the new oil was placed in it—seeming to prove that the old oil, when it had been used and had been distributed at that rate of speed, say 140 revolutions per minute through a period of 12 months, had improved in lubricating quality instead of being injured. That, perhaps, is a little more clear than as I mentioned it before. I think that is the point really that the paper wishes to come at—as to whether the lubricating quality of the oil is injured by use.

Prof. J. B Webb.—Mr. Almond has not stated where he thinks the half-pint of oil went to. It is possible that it evaporated, so that he had the good qualities of a pint and a half in a pint, and for that reason it operated better.

Mr. Geo. S. Strong.—I should like to mention a case in which I have known of similar lubrication. As a great many know, the old-fashioned river steam-boat engines carry a very high pressure, about 175 lbs. to 180 lbs., and the oil or lubricant which was formerly used in these engines was tallow. The very high temperature was necessarily very much against the use of such a lubricant, and a great many of the old steam-boat engineers, in breaking in a new engine, used beeswax and plumbago a certain length of time until the cylinder was thoroughly polished, and after that would use no lubricant whatever, excepting the lubricant which was furnished by the water of condensation. I know of an engine which was driven for nearly 10 years and never had a drop of oil put into the cylinder, either for valves or cylinder, and they never had any trouble with it. The cylinder was polished as nicely as a looking-glass inside. There were no scratches or indications of wear

whatever, showing that plumbago and beeswax had the effect of putting a polish on there which was very much harder than the cast-iron itself, and which made it unnecessary to lubricate those cylinders after they had that polish. I think that is so to a very large extent in locomotive practice to-day. After a cylinder has that polish the question of lubrication is not so important as the necessity for getting that polish in breaking in a new engine. So that in breaking in new engines now I always recommend for the first month that a great deal of plumbago and beeswax be used as a cylinder lubricant until that polish is obtained, and then the question of cylinder lubrication is not of so much importance.

*Prof. J. E. Denton.**—The endurance tests cited in the paper do not apply to the journals of a Thurston oil-tester alone, as Mr. Hawkins intimates. The conclusions are true for continuous rubbing between any two bearing surfaces with *no renewal of oil supply.* The various instances mentioned by speakers of the lubricating value of oil after continued service are all cases where the lubricant is never so far contaminated with foreign matter as to destroy its fluidity or reduce it to a pasty condition, as was the case in the experiments described with the small Thurston machine.

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* Author's Closure, under the Rules.

CCCCV.

THE EFFECTIVE AREA OF SCREWS.

BY D. S. JACOBUS, HOBOKEN, N. J.

(Member of the Society.)

THE following examples of measurements recently made of the performance of screws are regarded as valuable instances of the verification of the statement made in Rankine's *Shipbuilding*, p. 89, to wit: "The effective area is the sectional area already mentioned of the stream of water laid hold of by the propeller, and is generally, if not always, greater than the actual area, in a ratio which, in good, ordinary examples, is 1.2 or thereabouts, and is sometimes as high as 1.4, a fact probably due to the stiffness of the water, which communicates motion laterally amongst its particles."

The first instance is the case of a tug-boat upon which two screws were tried. These two screws were especially adapted for verifying the above theory, their actual disk area being the same, but their effective area greatly different. In one of the screws the blades were bent backward and had a form which tended to throw the water together after it was acted on, thus diminishing the effective area; the other was of the ordinary form, in which the centre lines of the blades are at right angles to the shaft, and in which the effective area is considerably greater than the actual.

In addition to these two tests the results obtained by Professor Denton in his trials of the ferry-boat *Bergen* will be made use of, as well as those of fair reliability obtained on the steamer *Homer Ramsdell.*

The ratio of the effective to the actual disk area of the screws was found to be as follows:

Tug-boat, with ordinary true-pitch screw		1.42
" " screw having blades projecting backward		.57
Ferry-boat *Bergen*, with ordinary true-pitch screw	at speed of 12.09 stat. miles per hour.	1.53
	" " 13.4 " " "	1.48
Steamer *Homer Ramsdell*, with ordinary true-pitch screw		1.20

The extremely low ratio for the screw with the blades projecting backward is probably not altogether due to the action of the blades in throwing the water together, because the loss produced by shock is taken the same as for the ordinary screw, whereas it is probably greater, on account of the fact that the blades were much thicker than in the screw that it replaced. In addition to this the area of each of the blades was about 25% smaller than in the other screw. All we can say in this case is, that a construction of screw which tends to diminish the effective area will increase the slip. In our present state of knowledge, we cannot predict exactly how much of the indicated horse power is effective at the wheel.

In estimating the thrust of the screw the effective horse-power is taken as equal to the indicated divided by 1.63, which figure was shown by Professor Rankine to be correct for the ordinary class of screw steamers. Although in some recent experiments this figure is found to be nearer 2 than 1.63, those made on a late date on American ships have given figures about the same as that given by Professor Rankine.

The dimensions of the wheels and data from which the above results are derived are:

	Bergen.		Tug-Boat. Ordin'y Screw.	Tug-Boat. Special Screw.	Homer Ramsdell.
Diameter of wheel in feet	8		8.5	8.5	11.1
Pitch of wheel in feet	8.9		12.0	14.0	15.8
Velocity of the boat through the water in statute miles per hour	12.09	13.40	11.77	11.32	16.02
Velocity of slip in statute miles per hour	2.18	2.88	1.73	4.5	2.00
Total indicated horse-power	435.4	684.7	342.6	329.5	1062.7
Estimated effective horse-power	267.1	420.1	210.2	202.1	652.0

The particular figures employed in the case of the ferry-boat *Bergen* are those for a single screw astern, and are an average of four tests, made from 11.55 A.M. to 1.15 P.M., September 28, 1889, and of two tests made later on the same day.

The methods employed in obtaining the data given above for the tug-boat screws were very thorough, and involve a method of estimating the velocity of the tides which appears to be as exact as making observations at the two ends of the course by means of floats, and in addition does not involve the extra labor

required if the latter method is employed. The tests were conducted in the following manner:

A course was selected on the Hudson River parallel to its bank. The ends of this course were taken at the points at which it crossed the centre lines of two streets of New York city which ran at right angles to the bank of the river. It was very easy to determine the exact time of crossing the centre lines of the two streets selected, for they approached the river on a grade which made it possible to sight along them for a considerable distance.

The boat was first brought up to its proper speed outside of the limits of the course, and, at the moment of crossing the centre line of the first street, the time was recorded, together with the reading of the log. Two persons kept an independent tally of the time reading to the nearest second. A third party read the log, and others took indicator cards. The speed was determined by a box counter, and by counting the total number of revolutions made during the runs.

Six runs, three in each direction, were made with each wheel, the separate results of which are given in Tables I. and II.

The method of estimating the tides consisted in laying off a curve of the velocities of the boat with reference to the shore for a given speed of the boat through the water, as shown in Figs. 217 and 218. Two curves are made, one for the runs going in one direction, and one for those going in the opposite direction. The ordinates of these curves represent the velocities, and the abscissæ the time of day at the middle portion of the run. One-half of the difference between the ordinates of the two curves corresponding to a given time will represent the velocity of the tide at that time.

There is a slight error involved in taking the average of the velocities of the tide during each run, but this for a short course is much less than the errors of observation of time, etc.

In order to determine the velocity with reference to the shore for a given speed of the boat through the water, it is necessary to run at as nearly a uniform speed as possible during the tests, and to correct for any slight variations which may occur. To make the small correction, assume, as a first approximation, that the velocity of the boat with reference to the shore, for the run in question, is proportional to the number of revolutions made by the screw, and determine the tides in the manner just de-

scribed. As a second approximation, make use of the tides determined by the first approximation in determining the correction which has to be made in the velocity with reference to the shore. Ordinarily the first approximation is all which it is necessary to make.

The tests of both wheels were taken at a time during which the tide changed, which, ordinarily, is the most unfavorable period that can be selected, if exact observations of slip are to

FIG. 217.

Curves for determining the Velocity of Tides.
Test of Special Screw.

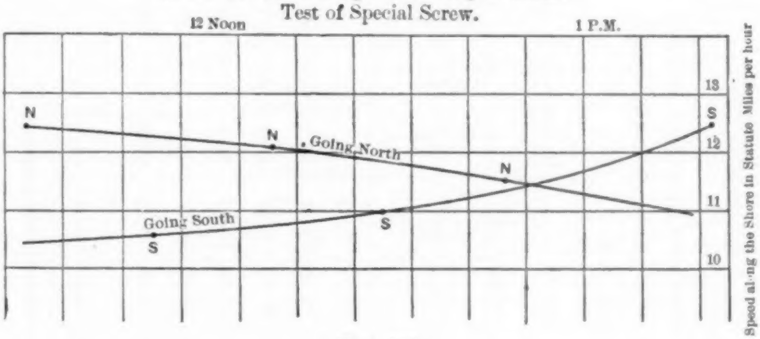
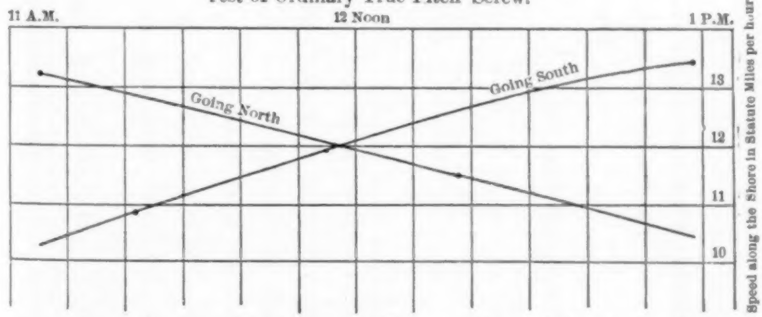


FIG. 218.

Curves for determining the Velocity of Tides.
Test of Ordinary True Pitch Screw.



be made. An examination of the results obtained for the per cent. of slip of the screws during the different runs will show that very uniform values were obtained by this method. Another check is obtained from the log. This could only be read to one-sixteenth of a knot, which involves a maximum probable error in a single run of 5%, or a maximum probable variation in two readings of 10%, and the maximum variation shown by the measurements is 11%, or just about the calculated amount.

TABLE I.

SCREW OF ORDINARY FORM HAVING TRUE PITCH. CENTRE LINE OF BLADES AT RIGHT ANGLES TO SHAFT.

$8\frac{1}{2}$ *feet diameter*, 12 *feet pitch*. *Length of course* = 2.72 *statute miles*.

Engine 18″ × 36″ × 26″.

Direction in which Boat runs.	Duration of Run in Minutes.	Revolutions of Screw per Minute.	Speeds in Statute Miles per Hour.				Slip of Screw in Per cent.	Log Readings.				Mean Effective Pressure.		Indicated Horse-Power.			Steam Pressure. Lbs. per Sq. In.		Vacuum in Inches of Mercury.
			Of Boat Relative to Shore.	Of Tide.*	True Speed of Boat.	Of Screw.		Amount Registered in Knots.	Speed per Hr. Knots.	Speed per Hr. Statute Miles.	Correction by which Reading of Log should be Divided.	High-Pressure Cylinder.	Low-Pressure Cylinder.	High-Pressure Cylinder.	Low-Pressure Cylinder.	Total.	At Boiler.	At Receiver.	
N....	12.25	101.0	13.32	+1.45	11.87	13.77	13.8	$2\frac{1}{4}$	11.02	12.71	1.07								
S....	15.00	100.8	10.88	+1.00	11.88	13.75	13.6	$2\frac{15}{16}$	11.75	13.55	1.14	54.7	13.3	181.9	177.5	359.4	93.0	17.0	27
N....	13.03	100.9	12.52	+ .55	11.97	13.76	13.0	$2\frac{1}{2}$	11.51	13.27	1.11	53.2	13.7	177.1	183.0	360.1	97.0	18.0	27
S....	14.25	95.7	11.45	+ .02	11.47	13.05	12.1	$2\frac{7}{16}$	10.27	11.84	1.03	48.3	11.6	152.5	147.0	299.5	83.3	12.6	$27\frac{1}{4}$
N....	14.83	96.2	11.00	− .55	11.55	13.12	12.0	$2\frac{3}{4}$	11.13	12.83	1.11	50.8	12.4	161.3	157.9	319.2	86.0	14.0	$27\frac{1}{4}$
S....	12.06	101.4	13.53	−1.55	11.98	13.83	13.4	$2\frac{1}{4}$	11.19	12.90	1.08	56.7	13.8	189.7	185.2	374.9	96.0	18.0	$26\frac{1}{2}$
Av. of Last Five Observations..		99.00	11.88		11.77	13.50	12.8	2.58	11.17	12.87	1.09	52.7	13.0	172.5	170.1	342.6	91.1	15.9	27.0

* If South or down stream, −. If North or up stream, +.

TABLE II.

SCREW WITH BLADES PROJECTING BACKWARD. $8\frac{1}{2}$ FEET DIAMETER, 14 FEET PITCH.

Length of Course = 2.72 *Statute Miles.*

Direction in which the Boat runs.	Duration of Run in Minutes.	Revolutions of Screw per Minute.	Speed in Statute Miles per Hour. Of Boat relative to Shore.	Speed in Statute Miles per Hour. Of Tide.*	Speed in Statute Miles per Hour. True Speed of Boat.	Speed in Statute Miles per Hour. Of Screw.	Slip of Screw in Per cent.	Log Readings. Amount Registered in Knots.	Log Readings. Speed per Hour. Knots.	Log Readings. Speed per Hour. Statute Miles.	Log Readings. Correction by which Reading of Log should be Divided.	Mean Effective Pressure. High-Pressure Cylinder.	Mean Effective Pressure. Low-Pressure Cylinder.	Indicated Horse-Power. High-Pressure Cylinder.	Indicated Horse-Power. Low-Pressure Cylinder.	Indicated Horse-Power. Total.	Steam Pressure. Lbs. per Sq. In. At Boiler.	Steam Pressure. Lbs. per Sq. In. At Receiver.	Vacuum in Inches of Mercury.
N	13.23	99.6	12.34	+1.00	11.34	15.85	28.5	$2\frac{3}{8}$	10.77	12.41	1.09	51.1	12.5	168.0	164.8	332.8	97	18	27
S	15.45	101.0	10.57	+ .90	11.47	16.07	28.6	$2\frac{13}{16}$	10.92	12.51	1.09	53.0	12.8	176.6	171.2	347.8	95	18	27
N	13.40	101.1	12.18	+ .70	11.48	16.08	28.6	$2\frac{7}{16}$	10.91	12.58	1.10	54.4	12.7	181.5	170.0	351.5	95	17	27
S	14.83	101.1	11.00	+ .45	11.45	16.08	28.8	$2\frac{3}{4}$	11.13	12.83	1.12	54.0	12.9	180.2	172.7	352.9	93	17	$26\frac{1}{2}$
N	14.37	99.2	11.36	+ .05	11.31	15.78	28.4	$2\frac{5}{8}$	10.96	12.63	1.12	50.0	11.4	163.7	149.8	313.5	94	17	27
S	13.97	94.6	11.68	− .80	10.88	15.05	27.7	$2\frac{3}{8}$	10.20	11.76	1.08	44.7	11.1	139.5	139.0	278.5	97.5	19	27
Average . . .		99.4	11.52		11.32	15.82	28.4	2.56	10.82	12.45	1.10	51.2	12.2	168.2	161.3	329.5	95.3	17.7	26.9

* If South, or down stream, −. If North, or up stream, +.

DISCUSSION.

Mr. Charles E. Emery.—This paper belongs to a class which should be encouraged, as it adds at least two more accurate experimental determinations of facts which will be valuable for reference in the final settlement of the now very unsatisfactory mathematical expressions in relation to marine propulsion. It is the province of the investigator carefully to ascertain and promulgate the results of experiments, and it is desirable, in such connection, to compare such results with those given by other experiments and accepted formulas. In this case, while complimenting the experiments, we must criticise the fact that the actual relative efficiencies of the two screws employed on the tug-boat are not presented. The special screw of the tug-boat shows considerably less than half the so-called "effective area" of the true screw by the Rankine rule and over double the slip, but the actual loss of efficiency is proportionally much less. In comparing actual results with the same hull and same engines, the efficiency practically varies as the cube of the actual speed divided by the indicated horse-power. Performing the operations, the result for the common screw is 4.7592 and for the special screw 4.4024, so that the common screw during this particular trial was only 8.11% more efficient than the other, provided several conditions not mentioned in the report were the same, more particularly the draught of water fixing the displacement of the hull. It is well known in practice that the pitch relative to diameter and the width of the blades may be varied within very considerable limits, so as to cause considerable variation in the "slip" and but little in efficiency. Thorneycroft obtained high efficiency with his "vortex" propeller, in which the vanes inclined backward very much; the builder of the phenomenal *Buzz* obtained his best efficiency with a wheel tending to disperse the water, and Herreschoff has found the same efficiency with the wheel put on backward as forward. As a general opinion, based on this conflict of evidence, many of us have come to believe that for wheels of the same diameter, with blades of even thickness and smoothness, the contour and direction of the blades have little influence on the result, though frequently the efficiency can be moderately increased by changing the widths of the blades. The narrow blades will always have a greater slip but less friction, so for changes within certain limits the efficiencies are not materially modified. The experiments under discussion confirm this view so

far as showing that neither Rankine's formulas nor the slip is a correct index of the efficiency. In this particular case the designer of the special propeller simply reduced his blades a little more than was proper for the particular conditions, but if the same machinery and propeller were put in a lighter vessel which would run at a greater speed with the same power, undoubtedly the special wheel would have shown a higher efficiency than the other, and on the contrary, for towing purposes the wider blades would probably show a higher relative efficiency than they did in this case. It is true that often very considerable gains in the efficiency of the machinery are made by changing propellers, and the firm which is successful in this respect always attributes it to the superior kind of propeller, whereas, in nine cases in ten, particularly with tug-boats, the propeller has simply been adapted to the boiler and engine. If a new boat comes out and steams freely, an expert can usually guarantee and obtain greater speed by putting in a propeller with less pitch which will permit the engine to work off the steam. The changes made in the propellers of such large steamers as the Inman *City of New York*, the White Star *Teutonic*, and in some of the German steamers, involve this principle. On the contrary, another new tugboat may come out in which it is difficult to keep up the steam pressure. The same expert will put on a new wheel with greater pitch, thereby reducing the revolutions, permitting the boiler to keep up the pressure, and thereby securing a better result as a whole, and another success is boasted of. The results in each case are based on the results obtained with the original wheels, and only those who are new in the business attempt improvements in propellers when the results already show a fairly good performance.

Mr. A. H. Raynal.—As a practical manufacturer of propeller wheels for some years, I take the liberty of calling the attention of the author, and of all those who look into the theoretical details of propellers, to the practical difficulty of measuring propeller wheels after they are made. All those who have made them heretofore will bear me out in this statement: from the moment that the mould is made until the propeller is put down on the table to be bored out there is a set of changes going on all the time, and you do not know at all what the actual pitch of your propeller is. More important than the shape of the propeller, no doubt, is the quality of the material of which the propeller is made, the degree of smoothness of the surface, and more than everything else, the sharpness of the edge. At the Delamater Iron Works we made propellers of all kinds of

pitches, and each inventor claimed wonderful results, and after all, the ordinary propeller stood there as well as any of them. But we did find a difference when we did not lay out the opposite blade, or when our propeller was out of balance, or the edges were poor or they were not properly cleaned. I only wish to state this so as to warn these gentlemen before they accept a certain pitch to make the most careful measurements of the propeller in question.

Mr. Oberlin Smith.—I think it would be interesting to bring out here any experiences in regard to the number of blades of propellers. I know that one of the new steamers—I think it was the *Majestic*—came over with three blades, where her sister ship had four. I believe the results with three were the better. We know that many boats run with two; and that question of the relative efficiency of two or three or four blades, other conditions being alike, is an interesting one.

Another important question in regard to propellers is what effect casings surrounding them may have on their action. If we accept the theory of Rankine's as to the effective area of push being the cross-section of the stream of water, that real cross-section may be different with and without a casing—probably somewhat larger than the propeller itself when not encased, I don't know exactly how much. With a casing, it would obviously be limited to the internal diameter of that casing. If anybody has experience on this point it would be interesting to have it brought before the meeting.

Mr. Gus. C. Henning.—I should like to say that one other source of error is found in modern propellers, especially in those of manganese bronze or other metals which permit of their being cast not more than half an inch thick at the edge with a blade which is several feet long. There is sufficient flexure in such blades to change the pitch totally, so that there is no more resemblance in the screw when under action and when at rest than if it were an entirely different one. That effect, I think, has never been taken into consideration at all.

Mr. J. W. Cole.—Has any of the gentlemen present any data as to propeller wheels run in cylindrical or conical cases, such as the speaker has mentioned?

Mr. H. H. Suplee.—I know of an elaborate series of experiments now in progress to determine that very point, but they have not been completed. If possible, I should be glad to answer the question if I can obtain access to the results.

Mr. Oberlin Smith.—I think the point brought out by Mr. Henning is a good one, and I would suggest that at some future meeting of this Society somebody bring a description of an apparatus for indicating the shape of a propeller wheel while in motion, showing the difference between its shape then and its shape at rest, and at other and different speeds. I have no doubt that such an apparatus could be contrived to show, approximately at least, whether there was much of that change of shape that Mr. Henning speaks of. It can, however, amount to a practical evil only upon extremely thin blades.

Prof. Jacobus.—I am glad that this paper has called forth so able a discussion by engineers having experience in this line, and especially thank Mr. Emery for his careful review of the matter.

In reply to the criticism that the efficiencies of the several screws are not included, I will say that they were worked out for the cases cited by Mr. Emery, but were not included, because it was thought best to limit the paper to a discussion of the effective areas. The draught of the tug-boat was the same in the two tests.

The pitch of the screws was measured after the blades were fastened to the hub, so that the actual amount is recorded.

CCCCVI.

INFLUENCE OF STEAM-JACKETS OF THE PAWTUCKET PUMPING ENGINE.

BY D. S. JACOBUS, HOBOKEN, N. J.

(Member of the Society.)

A SHORT time ago a paper was presented by Prof. Denton on the influence of steam-jackets on the economy and form of the indicator cards of the pumping engine at the Pawtucket Water Works. In this paper Prof. Denton remarked that the difference in the form of the indicator cards with and without jackets was so small that it was hard to draw any definite conclusion in regard to the actual amount. It was also stated that for this reason an arrangement was being made by which any error in the indicator drum motion would be eliminated, so that more definite conclusions could be arrived at. This work was handed over to me by Prof. Denton after several sets of cards had been taken with the new apparatus, both by himself and by Mr. Walker, the chief engineer of the works. Through the courtesy of Mr. Darling and Mr. Walker I was also presented with a set of data extending over a period of 21 days, during which time the engine was run continuously under three conditions, *viz.:* 1. Seven days with jackets and the reheater in use; 2. Two days with jackets shut off and flue reheater in use; 3. Five days with both jackets and flue reheater shut off. This series of data is very complete, readings of the important quantities being taken every hour, and in some cases every half hour.

The object of the present paper is to compare the law of expansion of the steam and economy obtained when working with and without jackets, and to show if the indicator cards taken by different parties at various times will lead to the same conclusions. With this end in view the writer took several sets of cards in person about one month after the time of making the tests previously quoted.

The arrangement used for communicating the motion to the

drum of the indicator shown in Fig. 221 has several advantages which will be fully explained in a description of the same.

The conclusions arrived at are: 1. The effect of jackets on the form of the expansion curves of both the high and low-pressure cylinders is so small that the errors involved in the most accurate measurements of the indicator diagrams make it impossible to show any difference in the laws governing the same; 2. The economy of the plant when working under the three sets of conditions is about 1.7% in favor of using the flue reheater, and 2.5% in favor of using both jackets and flue reheater over that obtained when neither the flue reheater or jackets are used. The duty for both jackets and flue heater in use was 121,580,000 ft.-lbs. per 100 lbs. of coal; 3. There was very little reëvaporation during expansion in either the high or low-pressure cylinder, the average for all the tests being .4% for the high cylinder, and 2.9% for the low.

Figs. 219 and 220 show the form of the indicator cards of the high and low-pressure cylinders.

Table I. contains the principal results derived in regard to the economy of the plant when working under the several sets of conditions. Tables II. and III. give the averages and final results of the measurements made on the indicator cards. Table IV. is inserted to show the way in which the data from each set of cards were derived. The number of cards in each set is 10 or 12.

In the case in which the jackets were in use—but not the heater—the reëvaporation in the low-pressure cylinder was greater than in any other case. As there is no apparent reason why the reëvaporation in this case should be different from that obtained when both the jackets and reheater are in use, and the result was not verified by making several tests, as in the other cases, it is not well to place too great stress on this figure.

Prof. Denton has already presented a description * of the engine to this Society, for which reason only the principal dimensions are given in Table I. It may be well to recall that the engine is a cross compound, and that both the high and low-pressure cylinders are jacketed on head and sides with steam of full boiler pressure.

It may be noticed that the steam per hour per horse-power calculated from the cards, and the percentage of steam at release in the high-pressure cylinder that is accounted for at cut-off in the

* See p. 328 of this volume.

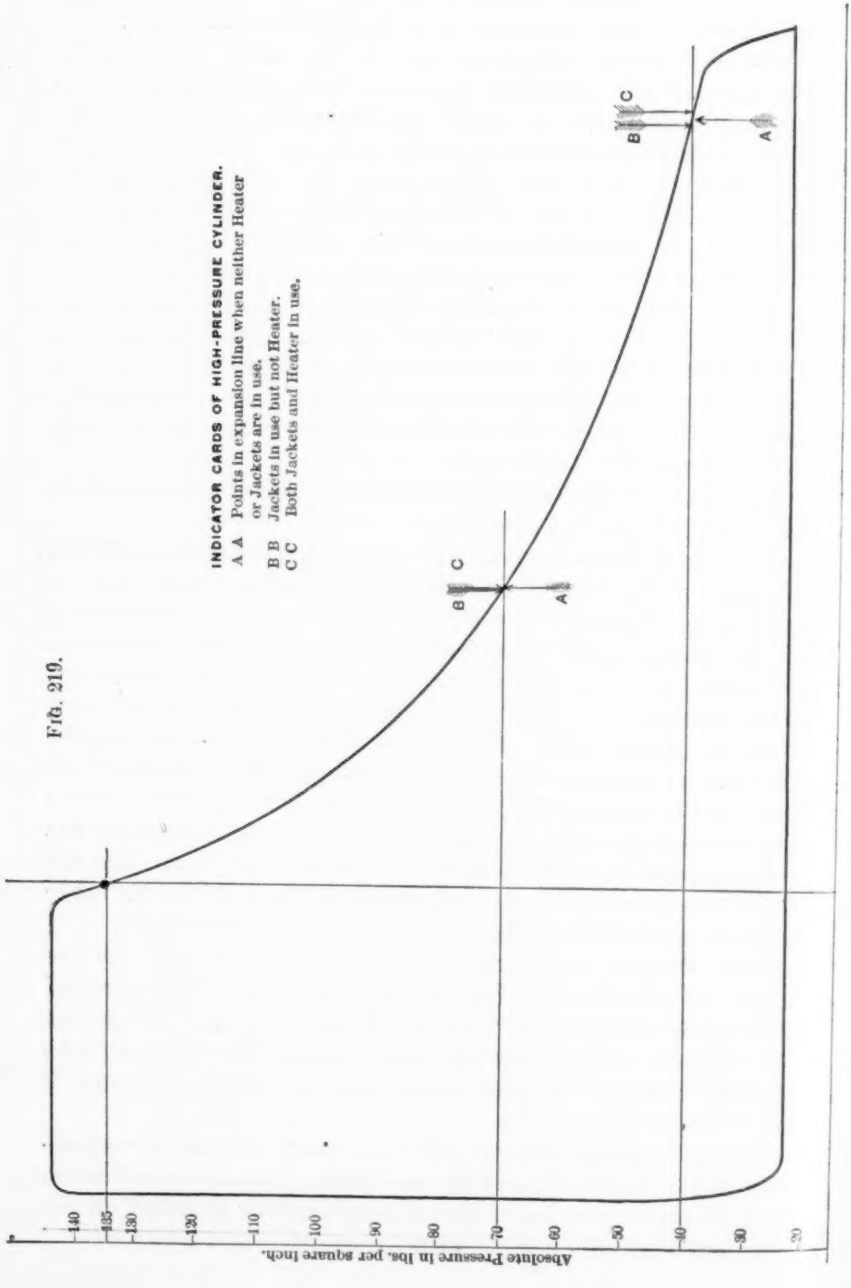

INDICATOR CARDS OF HIGH-PRESSURE CYLINDER.
A A Points in expansion line when neither Heater or Jackets are in use.
B B Jackets in use but not Heater.
C C Both Jackets and Heater in use.
A
B
C
Absolute Pressure in lbs. per square inch.
140
135
130
120
110
100
90
80
70
60
50
40
30
20

FIG. 219.

low-pressure cylinder, are not given in Table III.; these figures are not presented because the indicator cards were taken from one end only of the cylinders. It must not be inferred, because the law of expansion in the separate cylinders is the same when working with or without jackets, that the economy or combined diagram will be the same. To show the relative economy direct tests must be made by measuring the fuel or steam used, and to plot the combined diagram cards must be taken from both ends of the cylinders. At some future time exact cards may be taken from all the ends of the cylinders simultaneously, but for the purposes of the present article it was thought best to simplify matters in order to be able to discuss the law of expansion in each separate cylinder with as great accuracy as possible.

Fig. 221 represents the apparatus used to communicate motion to the drum of the indicator. It consists of a pantograph attached to the piston-rod, whereby the motion of the same is correctly reduced. A wooden rod is connected to the pantograph at the point at which the string leading to the indicator is usually attached; this is held by a guide so that it passes directly at the side of the indicator drum, as represented in the figure. The spring is removed from the indicator drum. To the drum are attached two catgut strings which wind partly around it and pass in opposite directions to violin keys placed in the wooden rod. The string is arranged so that it is exactly parallel to the centre line of the rod. By adjusting the violin keys the drum may be given the desired motion and the strings drawn very tight, so that there is no error due to the stretching of the same. No undue friction is caused on the drum spindle by tightening up the keys, for the tensions of the two cords balance each other.

The advantage of this arrangement is that it entirely does away with the stretching of the cord driving the drum, which, as modified by the friction of the drum and variation of the pull produced by the spring, causes slight errors in the card.*

A very slight inaccuracy in taking the indicator cards will produce an error which will more than cover the difference in the form of the cards under the several sets of conditions. For instance, on starting to take cards from the low-pressure cylinder the piston of the indicator leaked slightly and the pencil would not always travel over the same expansion line for several consecutive strokes. This was not due to the action of the governor,

* See article by Prof. Webb, p. 311 of this volume.

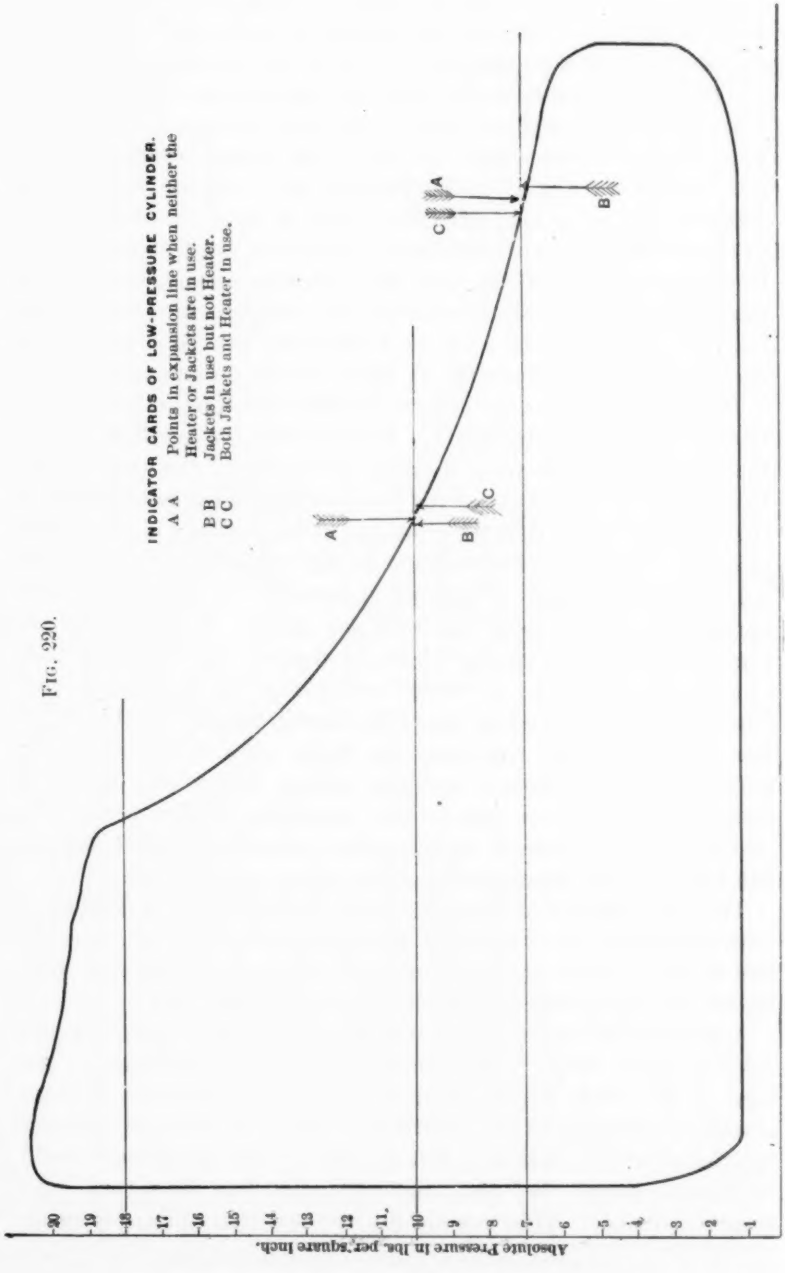

INDICATOR CARDS OF LOW-PRESSURE CYLINDER.
A A Points in expansion line when neither the Heater or Jackets are in use.
B B Jackets in use but not Heater.
C C Both Jackets and Heater in use.
A
B
C
Absolute Pressure in lbs. per square inch.
20 19 18 17 16 15 14 13 12 11 10 9 8 7 6 5 4 3 2 1

FIG. 220.

as it was clamped fast in taking some of the cards. On placing a better-fitting piston in the indicator, perfect diagrams were obtained. If a person, therefore, should have used the indicator with the slightly leaky piston, allowed the pencil to travel only one or two strokes and discarded those cards which had a double line on the supposition that it was caused by the governor, the errors involved would be much more than all the differences shown.

The pressure of the atmosphere, as shown by the barometer, was observed at the time that each set of cards was taken. If this had not been done the possible error in estimating the pressure shown by the low-pressure card would more than cover the difference found. For example, suppose the barometer to change one inch between the time that two sets of cards were taken; then the error in estimating the ratio of expansion shown by the low-pressure cards between the absolute pressures of 18 and 7 lbs. would be .09, whereas the maximum variation between the average results of tests made under the same conditions as given in Table II. is .08. The jackets were thoroughly blown out, and the engine ran for about four hours, or more, under each set of conditions before the indicator cards were taken.

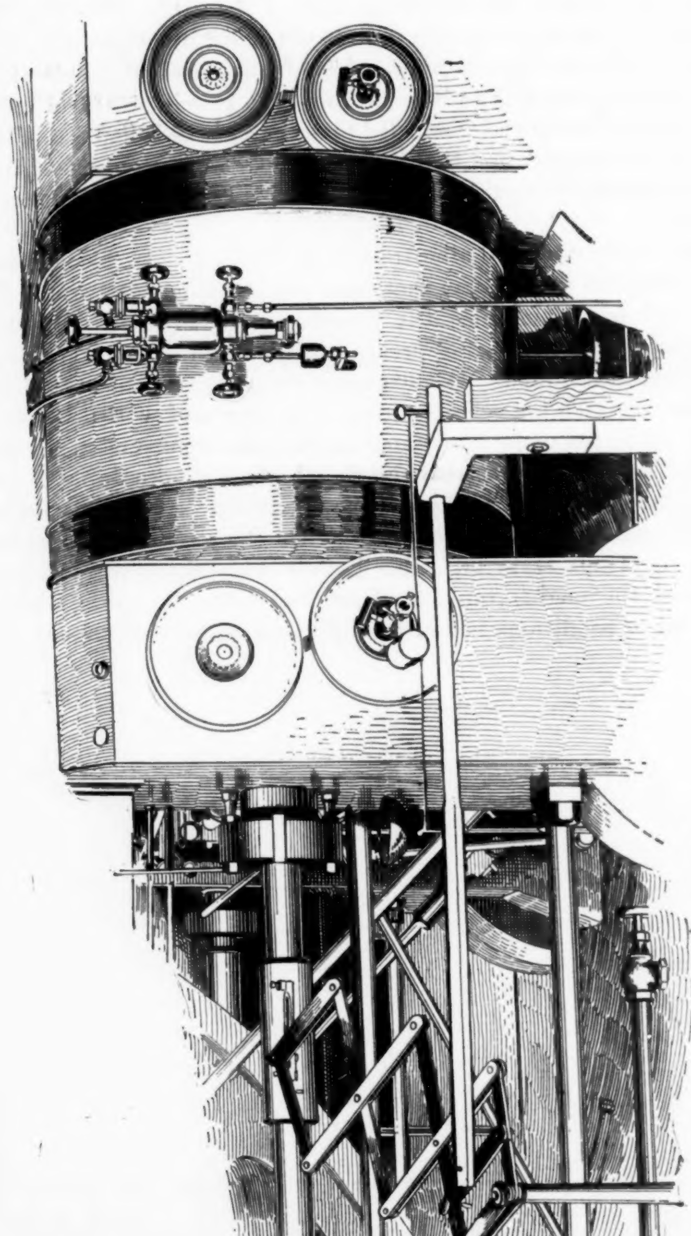


Fig. 221.

TABLE I.

TESTS OF ECONOMY OF THE PAWTUCKET PUMPING ENGINE.

Bore of Steam Cylinder, 15" and 30⅛"; Bore of Water Cylinder, 10.52"; Stroke of all Pistons, 30"; Clearance, High, 4%; Low, 3.7%; Diameter of Rods, 2¾".

	With Jackets and Flue Reheater.	With Flue Reheater but without Jackets.	Without Jackets or Flue Reheater.
Duration of test in hours	169.72	48.58	117.30
AVERAGE PRESSURES.			
Boiler in lbs. per sq. in. above atmos.	125.82	126.76	126.20
Water " " " " " " "	108.18	108.31	107.91
Receiver " " " " " " "	7.98	8.51	7.83
Vacuum in inches of mercury	28.22	28.08	28.13
Suction in feet and inches of water	13'10.5"	13'4.75"	12'11.9"
Barometrical pressure in inches	30.24	30.12	30.23
AVERAGE TEMPERATURES IN DEG. FAHR.			
Condensing water	35.39	34.53	35.81
Hot well	95.54	95.90	95.22
Feed-water	107.98	93.74	93.47
Steam in steam-pipe two feet from steam-chest	358.2	360.2	366.4
Temp. of saturated steam at boiler pressure	353.2	353.7	353.4
Degrees of superheating	5.0	6.5	13.0
Number of degrees that the temperature of the steam in the jackets of the low-pressure cylinder is greater than that due pressure of admission in the low cylinder	120		
Engine room	68.7	62.9	69.3
Atmosphere	28.7	29.2	33.8
TOTAL QUANTITIES.			
Number of revolutions	498494	144102	341067
Weight of coal burned in lbs	37910	11044	26441
CALCULATED QUANTITIES.			
Revolutions per minute	48.95	49.44	48.46
Work at water cylinder, ft.-lbs. per revolution	92464	92403	91936
Coal burned per revolution in lbs.	.076049	.076640	.077502
Duty in ft.-lbs. per 100 lbs. of coal	121,580,000	120,570,000	118,620,000

Ashes for entire run in lbs	9100
Clinkers " " " "	1511
Total coal for engine	75395
Coal for heating purposes in lbs	450
Coal for starting fires under boilers in lbs	997
Per cent. of ashes	11.8
" " clinkers	2.0

TABLE II.

Conditions.	Party by whom the Cards were taken.	Pressure of Steam in lbs. per sq. in. above the atmosphere.		Temperature of Steam in Degrees Fahrenheit on entering Steam-chest.	Vacuum in inches of Mercury.	Absolute Pressure in lbs. per sq. inch measured on Indicator Cards.								Ratio of Volume (including Clearance) to Volume swept through by Piston.				Ratio of Expansion.				Mean Effective Pressure.	
		Boiler.	Receiver.			At end of Compression.		Near Cut-off.		Near Middle of Expansion Curve.		Near Release.		Near Cut-off.		Near Release.		Near Middle of Expansion Curve.		Near Release.			
						H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.
Both Jackets and Flue Reheater in use.	Prof. Denton...	123		356.5	28.5	33	4.2	134	18	70	10	40	7	.284	.356	.893	.881	1.82	1.74	3.15	2.47	56.1	12.38
	Mr. Walker....	123	7.75	......	27.6	34	4.6	134	18	70	10	40	7	.291	.356	.920	.871	1.82	1.73	3.17	2.44	59.6	12.19
	"The Writer.".	125	9	360.3	27.7	35.5	4.5	134	18	70	10	40	7	.314	.387	.987	.955	1.82	1.77	3.15	2.47	62.5	13.05
	Average........	124	8.25	358.4	27.9	34.2	4.4	134	18	70	10	40	7	.296	.366	.933	.902	1.82	1.75	3.16	2.46	59.4	12.54
Jackets, but not Flue Reheater in use.	"The Writer.".	124.5	8.0	363.8	27.75	32.8	5.4	134	18	70	10	40	7	.324	.370	1.011	.936	1.81	1.72	3.12	2.53	66.3	11.30
Without Jackets and Flue Reheater in use.	Mr. Walker....	123	8	......	27.5	33	4.8	134	18	70	10	40	7	.317	.360	.997	.909	1.81	1.74	3.15	2.53	63.7	11.74
	"The Writer.".	125	8	363.8	27.9	34	4.5	134	18	70	10	40	7	.326	.372	1.014	.913	1.80	1.72	3.12	2.45	64.4	12.04
	Average........	124	8	363.8	27.7	33.5	4.7	134	18	70	10	40	7	.322	.366	1.006	.911	1.81	1.73	3.14	2.49	64.1	11.89

The special object of this table is to show how nearly cards taken by three different experts give the same law of expansion. The test to determine this is simply whether the values under the head of "Ratio of Expansion" are the same. For instance, the variation of this value for the low-pressure cylinder at release being between 2.53 and 2.44, the corresponding variation in the abscissæ on a card four inches long would be about .13″. Similarly the greatest variation for the high-pressure cylinder near release is between 3.12 and 3.17, and the corresponding variation in the abscissæ about .06″.

TABLE III.

Character of Test.	High and Low-Pressure Cylinder.	Ratio of Volume (including Clearance) to Volume swept through by Piston.			Absolute Pressures in Pounds per Square Inch.			Density of Steam in Pounds per Cubic Foot.			$A \times D - C \times F$.	$B \times E - C \times F$.	Reëvaporation during Expansion in per cent. of Steam accounted for, by Card, at Cut-off.
		Cut-off. A.	Release. B.	Clearance. C.	Cut-off.	Release.	End of Compression.	Cut-off. D.	Release. E.	End of Compression. F.			
With Jackets and Flue	H	.296	.933	.04	134	40	34.2	.3040	.0974	.0840	.0866	.0875	1.0
Reheater	L	.366	.902	.037	18	7	4.4	.0459	.0189	.0122	.0163	.0166	1.8
With Jackets, but with-	H	.324	1.011	.04	134	40	32.8	.3040	.0974	.0867	.0953	.0953	.0
out Flue Reheater ..	L	.370	.936	.037	18	7	5.4	.0459	.0189	.0148	.0164	.0171	4.3
Without Jackets and	H	.322	1.006	.04	134	40	33.5	.3040	.0974	.0824	.0946	.0947	.1
Flue Reheater......	L	.366	.911	.037	18	7	4.7	.0459	.0189	.0130	.0163	.0167	2.5

The object of this table is to show the variation in reëvaporation, which is given in the last column.

TABLE IV.

WITH JACKETS AND FLUE REHEATER IN USE.

Barometer, 29.55″ *Revolutions per minute*, 47.6.

Time.	Pressure of Steam above the Atmosphere.		Temperature of Steam in Degrees Fahrenheit on entering Steam-chest.	Vacuum in inches of Mercury.	Absolute Pressures laid off on Indicator Cards.						Distances from Admission End of Indicator Cards in inches.								Ratio of Volume (including Clearance) to Volume swept through by Piston.						Ratio of Expansion.				Mean Effective Pressures.		Absolute Pressures at end of Compression.	
	Boiler.	Receiver.			Near Cut-off.		Near Middle of Expansion Curve.		Near Release.		Length.		Point near Cut-off.		Point near Middle of Expansion Curve.		Point near Release.		Point near Cut-off.		Point near Middle of Expansion Curve.		Point near Release.		Point at Middle of Expansion Curve.		Point near Release.					
					H. P.	L. P.	H. P	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.	H. P.	L. P.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33
P. M. 4.07	125	9.0	362.5	27.7	134	18	70	10	40	7	4.84	5.03	1.31	1.76	2.54	3.26	4.57	4.71	.311	.387	.565	685	.984	.973	1.82	1.77	3.16	2.52	62.4	12.8	35.0	4.7
4.10	125	9.0	362.0	27.7	134	18	70	10	40	7	4.84	5.03	1.32	1.75	2.54	3.27	4.55	4.67	.313	.385	.565	.687	.980	.965	1.81	1.78	3.13	2.51	62.4	13.2	36.0	4.5
4.11	125	9.0	361.0	27.7	134	18	70	10	40	7	4.84	5.03	1.36	1.77	2.62	3.26	4.64	4.61	.321	.389	.581	.685	.999	.953	1.81	1.76	3.11	2.45	63.2	13.1	36.0	4.5
4.12	125	9.0	360.0	27.7	134	18	70	10	40	7	4.84	5.03	1.33	1.77	2.58	3.27	4.61	4.58	.315	.389	.573	.687	.992	.948	1.82	1.77	3.15	2.44	63.2	13.1	35.0	4.5
4.14	125	9.0	358.5	27.8	134	18	70	10	40	7	4.84	5.03	1.33	1.76	2.58	3.23	4.62	4.58	.315	.387	.573	.679	.994	.948	1.82	1.76	3.16	2.45	63.2	13.1	35.0	4.5
4.16	125	9.0	358 0	27.8	134	18	70	10	40	7	4.84	5.03	1.30	1.74	2.53	3.22	4.52	4.55	.308	.383	.563	.677	.974	.942	1.83	1.77	3.16	2.46	60.8	13.0	36.0	4.2
Averages.....	125	9.0	360.3	27.7	134	18	70	10	40	7	4.84	5.03	1.33	1.76	2.57	3.25	4.59	4.62	.314	.387	.570	.683	.987	.955	1.82	1.77	3.15	2.47	62.5	13.1	35.5	4.5

The object of this table is simply to illustrate in detail the steps of the method pursued in obtaining the ratio of expansion and reëvaporation.

DISCUSSION.

Mr. A. F. Nagle.—In reading this paper we shall certainly be misled if we overlook the fact that the steam used in this engine was superheated 5° when the test was made with steam in the jackets, and 13° without boiler steam in the jackets, or a flue heater.

The slight gain by the jacket in the first cylinder of this engine does not justify the extra cost and hazard incurred, and confirms the position taken by Mr. Corliss in his mill-engine practice. All his mill engines were without jackets, and he claimed to have demonstrated their needlessness when the engineering practice of the world was establishing their value. I do not know personally how thoroughly Mr. Corliss had made his investigations, but his results justified his conclusions. But it should be known that Mr. Corliss always used a vertical tubular boiler with a large exposure of surface for superheating the steam, and it is due to this dry or superheated steam that he found no gain by jackets. With other boilers giving moist steam, the decided advantages by jacketing have been established beyond all doubt.

I am somewhat sceptical as to the correctness of the "vacuum in inches of mercury = 28.22;" "barometrical pressure in inches = 30.24."

The natural impression is that the vacuum was measured by a mercurial column, but as a fact it is measured by a Bourdon pressure gauge, and unless the gauge is verified I should not believe such an excellent vacuum attainable in a steam-engine. The habit of speaking of the vacuum in inches of mercury is bad and misleading and should not be indulged in unless it is really true.

Mr. H. H. Suplee.—In regard to Mr. Nagle's last remark, I had some time ago occasion to doubt the perfection of the vacuum in a compound Corliss engine as shown by the vacuum gauge, and inserted a mercury column to check the results. I found that a vacuum of a little over $28\frac{1}{10}$ inches by gauge was fully borne out by the mercury column inserted in the pipe at the same point as the Bourdon gauge. But when the barometer was only a little over 30 inches, I had doubted, as Mr. Nagle had doubted, the accuracy of the vacuum gauge when showing such excellent results.

Mr. A. H. Raynal.—During early experiments with the Ericsson expansion engine I found it impossible to obtain on the card

the same vacuum as in the cylinder. I got the best indicators I could get hold of, among others those of the Ashcroft Manufacturing Company, and I never reached closer than 2 inches of the actual vacuum in the cylinder. Of course it is very plain why, because the piston has to be to a certain extent leaky, and it takes but a very small amount of air to destroy that vacuum. We had within the cylinder a vacuum coming within one-half inch of an absolute vacuum, and the best, as I said before, we could obtain on the card by the indicator was usually out about 2 inches. We had in the other part of the cylinder a gauge attached which showed the accurate vacuum, and doubting it, we obtained other gauges, and correcting that and doubting these other gauges again, we put in the mercury column, and found we had the absolute vacuum within half an inch.

Mr. C. W. Barnaby.—I just want to refer to the statement that there is no error due to the stretching of the cords attached in the manner described. I think that would only be strictly true in the case of a cord which was entirely inelastic and where there was no inertia of the drum. That is, the motion of the indicator drum would coincide with the motion of the driving-stick referred to only in the case of an inelastic cord and in the case of entire absence of inertia of the drum and of elasticity of the cord. The error, however, in practice would be hardly perceptible.

Prof. J. B. Webb.—I wish to emphasize a point which Prof. Jacobus has not, perhaps, sufficiently dwelt upon. The paper says the spring is removed from the indicator drum; the removal of the spring is mainly to get rid of the drum friction. As far as the elasticity of the string is concerned it may lengthen the card, but it cannot produce any sensible error, so that there is no object in getting rid of it entirely. The elasticity of the string simply lengthens the card, leaving its proportions the same.

Mr. Jas. W. See.—It seems to me that the device mentioned by Mr. Jacobus is a practically perfect one, but I recognize it merely as a device for converting reciprocating motion into rotary motion. But let me ask what advantage it has over a rack and pinion, and why a rack and pinion of a fair practical construction and without material lost motion would not be a practical device to put on all indicators, and give a man a chance to use his positive motion or string motion as he desires?

Mr. Daniel Ashworth.—In referring to indicators, we are all well aware that it is the usual practice in the lazy-tongs attachment to

place the posts mid-stroke, and hence we get a diagonal movement, the accuracy of which is doubtful, and it is also very difficult to erect and place in one position except in an engine room where the floor is properly prepared for it. For quite a long while I have followed the practice of taking a hold direct upon the lazy-tongs, clasping a nut on the cylinder-head and pulling it straight out coincident with the piston. The whole matter is so simple that you can carry all the attachments almost in your pocket, dropping it on in an instant and taking it off. I have no difficulty in pulling straight out and closing straight up coincident with the engine and parallel to it. If you have the post system of placing it at mid-stroke your cylinder may be in such a position that your line will reach for the lazy-tongs at an angle, but in this case all this is avoided and you have exceedingly short lines.

Mr. H. H. Suplee.—Leaving for a moment this question of indicator motions and returning to the question of defective vacuum, I have found the difficulty of obtaining anything like the vacuum on the cards that the gauge shows, and where I have verified the vacuum with a mercury gauge I took pains to attach the mercury column at successive points between the condenser and low-pressure cylinder, attaching it to a pipe which has a moderate length, perhaps six or seven feet, and then attaching it to the cylinder also, and I found that the loss in vacuum was almost graduated, continually falling between the condenser and cylinder as if it was due to some other resistance than merely the loss in the instrument itself.

Prof. Jacobus.—In reply to Mr. Nagle's criticism in regard to the vacuum gauge, I will say that the only back-pressure which affects the economy is that shown by the indicators, the springs of which were specially made for the test and certified to by the manufacturers. The reading of the vacuum gauge is included simply to give all the conditions which existed during the test. The gauge showed the back-pressure during the exhaust of the low-pressure cylinder to be about one-half a pound per square inch, a result which appears to be about correct. The statement that the effect of jackets is so much more beneficial with wet than with dry steam, or steam which is slightly superheated, is interesting, and I wish Mr. Nagle would advise us in regard to the particular tests from which he draws this conclusion.

We have tried the method which Mr. Ashworth suggests of holding the lazy-tongs at one end and attaching the other to the

cross-head so that it is drawn in and out in a line parallel to the movement of the piston, but have experienced difficulty in working it in this way throughout a long test on account of the great amount of movement which necessarily occurs at the joints. When attached in this way to an engine having a long stroke and high speed the strain is very great. In one case which came under our observation the fastenings at the ends of the lazy-tongs gave way. If one end is held at or near the centre of the stroke the amount of movement at the joints is reduced and there is less strain on the same.

ADDED SINCE THE MEETING.

As the accuracy of the vacuum gauge has been questioned, and as this is the only instrument employed in the tests which was not calibrated, it was determined to compare its readings with a mercury column. The column was taken to the pumping station and compared with the gauge without removing it from the position it occupied during the test. It was found that the gauge gave too high a reading by 0.65 inch of mercury, or the error of reading was about one-third of a pound per square inch. In other words, 0.65 inch should be subtracted from the readings in the table in order to obtain the true vacuum.

CCCCVII.

THE EFFECT OF AN UNBALANCED ECCENTRIC OR GOVERNOR BALL ON THE VALVE MOTION OF SHAFT-GOVERNED ENGINES.

BY JOHN E. SWEET, SYRACUSE, N. Y.

(Member of the Society, and Past President.)

THE fast-growing family of steam-engine governors, known as shaft governors, can be divided into two classes. In one class, the eccentric is forced over, through the agency of some wedging action, such as a secondary eccentric or slide. In the other class, the eccentric is pulled or pushed, by the force of the governing weight direct.

The advantage of the first class is, that any resistance in the valve does not recoil on the governing weight, and hence affect it, and the disadvantage is, that the extra friction impairs its sensitiveness. The advantage in the second class is that of possible sensitive governing: and the disadvantage of being easily disturbed by external influences.

In discussing the question some ten years ago, with the late John Coffin, as to what would be the effect on the valve motion of an engine, were the governor ball unbalanced, he said, "The ball would revolve in a circle, but the centre of the circle would be below the centre of the shaft."*

This seemed so reasonable that it was accepted as a fact—until the last year we labored under the delusion, notwithstanding that disturbed valve motions were occurring with disagreeable frequency. We contented ourselves by attributing the cause to every possible agency but the true one. Something like a year ago, Mr. E. J. Armstrong, of this Society, had an aggravated case of valve disturbance in an engine well suited to experimenting; and as a matter of investigation, he fixed a pencil

* This is the only case of the kind where I ever knew him to be mistaken, and the last letter I ever received from him was in acknowledgment of this error.—J. E. S.

in the eccentric and obtained the diagram, accurately reproduced in Fig. 178.

His belief that the distortion of the eccentric curve must have been caused by the unbalanced eccentric, led him to the consideration of the effect of gravity on a weight, working under the condition of an unbalanced governor ball. For the action of an unbalanced eccentric is the same in extent as a governor ball which had added to it weight enough to counterbalance the eccentric, and with the eccentric omitted.

To put this problem in its most simple form, what will be the effect of gravity on an unbalanced governor weight? Assume that the ordinary fly-wheel *A*, Fig. 179, has arranged within it a weight *B*, free to move radially, and held in, against centrifu-

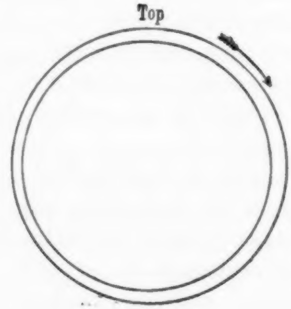


FIG. 178.

gal force, by the spring *C*. With the spring properly proportioned to a given speed, given the proper initial tension, and the wheel maintained at that speed, the tendency of the ball to go out by centrifugal force and the tendency of the spring to draw it in, will exactly balance each other, wheresoever the ball may be placed. So that, were it not for the disturbing influence of gravity, the ball might be set anywhere out or in, and would there stay, so long as the constant speed of the wheel was maintained. To consider what the effect of gravity would be, acting upon this weight under these conditions, assume, when the wheel is turning in the direction shown by the arrow, that the weight is at *a*. When the weight has reached the position *b*, gravity has drawn it out from the centre: at *c*, it is farther out: at *d*, farther yet: at *e*, still farther, and so on until it arrives at *g*, after which, at *h*, it is drawn in by gravity, and so on would reach the point *a*, at the completion of the revolution. This curved

path of the disturbed ball corresponds with the eccentric curve obtained in Armstrong's experiment; and it will be seen that, instead of being a circle, with its centre below the centre of the shaft, it is not a circle, and the centre is directly at one side, in the best possible position to disturb the valve motion of

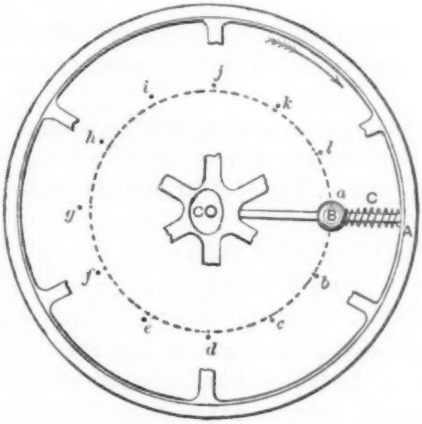


FIG. 179.

a horizontal engine. It will also by a slight reflection be understood, that the slower the speed, the greater the disturbance; because the longer the time given to gravity to act, the greater its effect.

I do not know that the discovery of this fact in regard to the revolving weight, or this method of presenting it, is here given for the first time, but if it is, Mr. Armstrong is entitled to due credit for its discovery and its elucidation.

The effect of this disturbance, caused by either an unbalanced eccentric, governor ball, or other unbalanced elements in the governing mechanism, will be better understood by reference to the small curved line *C*, which is parallel to the path of the ball.

DISCUSSION.

Prof. S. W. Robinson.—In my class-room lectures I have given mathematical analysis of this case of an unbalanced governor weight, such as a single weight revolving about a horizontal shaft and free to move out against a spring resistance.

The result of this analysis was found to be that the weight would move in a curve nearly a circle, the centre of which is raised exactly upward to a height above the centre of the shaft by an amount

$$h = \frac{g}{\omega^2} = \frac{g}{4\pi^2 n^2},$$

where g is the acceleration of gravity $= 32.2$, $\omega =$ the angular velocity of revolution, and $n =$ the number of revolutions per second.

This result of analysis agreed with what seemed to me to be a fair conclusion drawn from an inspection of the geometrical figure of the case, such as Fig. 179 of Prof. Sweet's paper, with reasoning about as follows:

Suppose, to start with, that the weight B is at j, moving along a horizontal portion of arc. The action of gravity tends to deflect it downward instead of allowing it to move along a circle path concentric with the shaft, thus giving the mass B an accelerated motion relative to the wheel A and along the radius AB, so that by the time the weight reaches a it will have a considerable velocity toward the centre C. From this point on gravity counteracts, and on reaching d will have destroyed the radial velocity toward C, when B will again be moving horizontally, or perpendicular to the radius, but will be at a point nearer the centre C than when at j. Now, from this point on a radial acceleration will occur, so that at g the weight will be moving outward with a radial velocity which from g on to j will again be destroyed by gravity, thus bringing the weight to rest on the radius at j, though at a greater radial distance from the centre C than at any point before in the revolution, and putting the weight in the position and condition supposed at the start, when it will go on in repetition of the curve as the next turn of the shaft is made, and so on continuously, the curved path described being found to be nearly a circle with its centre elevated above that of the shaft.

This reasoning around the circle while the spring is the only force acting on the weight besides gravity will not readily give any other conclusion than the above.

This corroboration of the result of analysis satisfied me until I was startled on noting the testimony of Prof. Sweet's paper, whereupon I determined to make some experimental trials of the matter.

To this end I rigged up a disk carrying a centrifugal device shown in Fig. 232, the affair being put into nearly isochronous condition by arranging to have the spring free from tension when the moving end was a little above the intersection of the line of the spring with the line *DE*. A pencil was mounted in the lever at *P*.

When motion was given to the disk, the centrifugal force balanced the spring at the angular velocity $\omega = 56.86$, or 555 revolutions

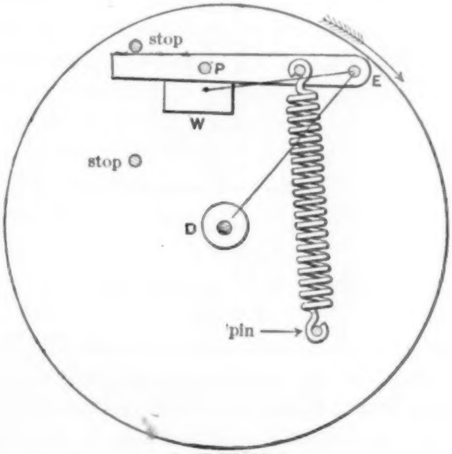


FIG. 232.

per minute, according to the determination from the weight in pounds of *W* and the lever, the reaction of the spring, etc., all quantities of data being carefully measured.

While the device was revolving right-handed, with the weight between stops, a board upon which a sheet of paper was tacked, and so placed as to swing to or from the pencil, was moved against the pencil and a curve obtained. This was done repeatedly. Also the circles of motion were got for the weight against the inner stop and the outer stop.

Examining the curves thus obtained with dividers, no appreciable deviation from the circular form was noticeable, and the centres of 20 such circles were read off in height above and deviation to right or left and noted. The results are given in the following

TABLE OF DISPLACEMENT OF CENTRE OF MOTION OF A SINGLE UNBALANCED GOVERNOR WEIGHT.

Remarks.	Centre Shifted Up, inches.	Centre Shifted to Right, inches.
Pencil light and	0.10″	0.00
Steady circle.	0.10″	0.00
Spiral, in, $\frac{3}{16}$″	0.13″	0.05
" " $\frac{1}{16}$	0.15	0.02
" " $\frac{1}{16}$	0.14	0.02
Spiral, slightly Steady (Pencil heavy)	0.035	0.035
" (Pencil heavy)	0.14	0.03
Spiral, out, $\frac{3}{32}$ (Pencil heavy)	0.14	0.03
" " $\frac{1}{16}$ (Pencil heavy)	0.12	0.02
" " $\frac{1}{8}$ (Pencil heavy)	0.12	0.03
" " $\frac{1}{8}$ (Pencil heavy)	0.13	0.02
Steady	0.05	0.00
"	0.16	0.02
"	0.16	0.02
"	0.16	0.02
"	0.15	0.013
Spiral, out, $\frac{3}{16}$	0.12	0.04
" in, $\frac{1}{16}$	0.15	0.04
" " $\frac{1}{8}$	0.10	0.037
Means	0.1257	0.0233

It is to be noted that the centres were slightly shifted to the right while the motion was left-handed relative to the diagrams. This may probably be accounted for by the fact of the slight friction of the lever on its fulcrum, as increased by the side pressure from the pencil. This conclusion is strengthened from the fact that the first two curves read give zero lateral shift, while the same curves are noted as having light pencil pressure, and they were in fact the lightest and probably the fairest results, in this respect, of all taken.

It may be possible that considerable disturbing forces may cause larger lateral displacement, but it is evident from the table that the displacement is mostly in the vertical direction, thus decidedly favoring the results of analysis and of reasoning upon the figure.

To make a crucial test of the analysis, we may take the value $0.1257'' = h$ and determine ω from the analytical result. Reducing 0.1257 to feet,

$$h = \frac{g}{\omega^2},$$

whence

$$\omega^2 = \frac{32.2 \times 12}{0.1257},$$

or

$$\omega = 55.26;$$

a very close agreement with the result 56.86 found by centrifugal force, spring tension, etc., of the device, results which are at once seen to be from entirely different principles and different data, and furnish a third and very strong attestation to the *vertical* instead of *horizontal* displacement of the unbalanced governor weight.

How the experiments reported by Prof. Sweet could have given a horizontal instead of a vertical displacement is a question it may be difficult to answer now from the results reported, and I would raise the question for Professor Sweet's consideration whether the apparatus used by him was not largely in error of isochronism, or whether there were not other forces acting to influence the weight besides the spring, centrifugal force, and gravity.

It may be remarked that in designing single-weight governors for horizontal shafts the displacement of the centre of the circle of motion may be found from the formula given, $h\,\omega^2 = g$, which, by reducing, gives for h in inches and number of revolutions per second,

$$h'' = \frac{9.778}{n^2}.$$

By aid of this formula the following table is given :

TABLE OF VERTICAL DISPLACEMENT OF A SINGLE UNBALANCED GOVERNOR WEIGHT.

Revolutions per Second.	Revolutions per Minute.	Vertical Displacement.
1	60	9.78 in.
2	120	2.44 "
3	180	1.09 "
4	240	0.61 "
5	300	0.39 "
6	360	0.27 "
8	480	0.15 "
10	600	0.09 "
12	720	0.07 "
14	840	0.05 "
16	960	0.04 "
18	1,080	0.03 "
20	1,200	0.02 "

According to this table one would hardly propose an unbalanced governor for a speed less than 200 revolutions per minute.

It is interesting to note that the result of the vertical displacement of the unbalanced single governor weight, viz.,

$$h = \frac{g}{\omega^2},$$

is the samé identically as is found for the height of the centre of the circles of curvature of the water in the buckets of the bucket water-wheel, known also as the breast wheel and overshot wheel, where the water in the buckets of the revolving wheel is, as is well known by hydraulic engineers, inclined outward by the centrifugal force due to the motion of the wheel to a cylindrical concavity of surface; all cylinders of concavity for the different buckets being, at a given instant, at a common point at the height $h = \frac{g}{\omega^2}$ above the axis of the wheel.

The analysis corroborating the results of the experiments cited in this discussion is as follows: In Fig. 233 let S be the shaft of the shaft governor, T a pivot in the wheel about which the weight W, fast on a lever, WT, swings in a path which for convenience is taken practically radial with respect to S, and U a spring counteracting centrifugal force. Let the initial position of W be directly over the shaft, and denote any angle between that and any other position by φ, as shown. Now, the only motion of W relative to the wheel will be outward and inward to and from S.

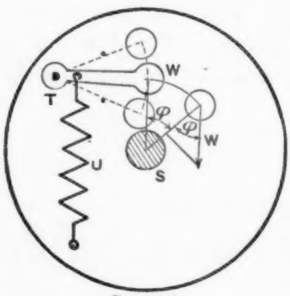


Fig. 233.

The forces acting on W will be the centrifugal force F, the counteracting force of the spring $= p\,(r - r_0)$, in which r is the distance of the weight W from S and r_0 the distance of W from the point on the line SW where the spring tension would be zero if the law of action of spring were to hold, as on the line SW, to the extent r_0. The disturbing influence from the valve is omitted.

If r_1 be the distance of weight from S for the engine running in regulated condition, with W between stops, then in the absence of gravity, as if the shaft were vertical, the forces acting on the weight would be, if F is the centrifugal force of the weight W and p the unit spring action,

$$F - p\,(r_1 - r_0),$$

and for equilibrium this $= O \quad \therefore \quad p = \frac{F}{r_1 - r_0} = \frac{W}{g}\,\frac{\omega^2 r_1}{r_1 - r_0}.$

But with the shaft in a horizontal position and W in the posi-

tion φ shown, the component of gravity, $W \cos \varphi$ must be subtracted from the expression of force, giving

$$F - p(r - r_0) - W \cos \varphi$$
$$= \frac{W}{g} \omega^2 r - \frac{W}{g} \frac{\omega^2 r_1}{r_1 - r_0} (r - r_0) - W \cos \varphi.$$

Now, this force is not constant during a revolution, but varies because $W \cos \varphi$ acts toward the centre S for the upper half of the revolution and from the centre for the lower half. Then the weight W being supposed free to move, it will be swung back and forth from S in a sort of vibratory motion of acceleration and retardation as the governor revolves. We have, then, only to put this force equal the differential expression for acceleration, viz.:

$$\frac{W}{g} \frac{d^2 r}{dt^2} = \frac{W}{g} \omega^2 r - \frac{W}{g} \frac{\omega^2 r_1}{r_1 - r_0} (r - r_0) - W \cos \varphi$$

or
$$\frac{d^2 r}{dt^2} = \omega^2 r - \frac{\omega^2 r_1}{r_1 - r_0} (r - r_0) - g \cos \varphi$$

$$= \frac{\omega^2 r_1 r_0}{r_1 - r_0} - \frac{\omega^2 r_0 r}{r_1 - r_0} - g \cos \omega t$$

φ being $= \omega t$ where ω is the angular velocity of wheel and t the time.

This general equation can be integrated by La Grange's method of the variation of parameters, and it leads to an expression of r as a trigonometric function of φ, or of ωt, an equation which adds great interest to the question of the motion of an unbalanced governor weight, showing that when the governor is not isochronous the weight may describe curves reëntering only after several revolutions, and hence giving a motion that might readily be taken as irregular as far as a governor for engines is concerned.

Leaving the full investigation for a future paper, let us suppose the governor isochronous, in which case $r_0 = O$. Then the first two terms of the second member vanish, reducing the equation to

$$\frac{d^2 r}{dt^2} = - g \cos \omega t.$$

Integrating once gives

$$\frac{dr}{dt} = -\frac{g}{\omega} \sin \omega t + C.$$

If we take the path as horizontal at the top where $\varphi = O$, then $t = O$ and the first member is zero also, so that $C = O$.

Integrating again we obtain

$$r = + \frac{g}{\omega^2} \cos \omega t + C'.$$

Designating by r' the radius to the top point of the curve where $t = O$, we obtain

$$r' = + \frac{g}{\omega^2} + C'.$$

Eliminating C' we get

$$r - r' = \frac{g}{\omega^2} (\cos \varphi - 1)$$

for the polar equation of the path, or of the orbit described by the governor weight.

At the lowest point of the curve $\varphi = \pi$, and designating by r'' the particular radius to this point, we obtain

$$r' - r'' = \frac{2g}{\omega^2},$$

in which r' must be greater than r'', placing the middle point between the top and bottom points of the orbit at a height $\frac{1}{2}(r' - r'')$ *above* the centre of the shaft, or at a height $\frac{g}{\omega^2}$, as stated in the discussion above.

In the last equation we get

$$\frac{g}{\omega^2} = \frac{r' - r''}{2},$$

which, substituted in the equation of the orbit, gives

$$r + \frac{r' + r''}{2} = \frac{r' - r''}{2} \cos \varphi,$$

or,

$$r + r_1 = \frac{r' - r''}{2} \cos \varphi,$$

if we call r_1 the mean of the two extreme radii r' and r''.

In this equation, if $\varphi = \frac{\pi}{2} = 90° \; r = r_1$, the mean radius, and the radius of the circular path in the absence of gravity as proved by

the first equation of acceleration, in which for this case the component of gravity $W \cos \varphi = O$, and also the accelerating force $= O$, giving $r = r_1$.

The orbit, as seen by the last equation above, is the curve of cosines where the base is a semicircle.

To illustrate by aid of a diagram (Fig. 234), take O for the main shaft of the governor, the dotted circle $ACBF$ to radius r_1 for the orbit, in the absence of gravity. Then the orbit for the unbalanced weight in the presence of gravity is the full line $ADBE$, where $CD = EF = r' - r_1 = r_1 - r''$; $GH = IJ = r - r_1$ above, or $r_1 - r$ below, etc., the portion ADB being the curve of cosines for ACB, the base, and AEB the curve of cosines for AEB as base.

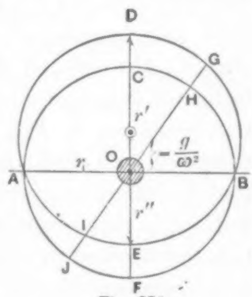


Fig. 234.

In the general integral of the complete expression above, it is shown that for the isochronous governor the orbit can only be horizontal at top and bottom; so that this curve of Fig. 234 must persist and give a regular governor, while if it is not isochronous the orbit is a multiplex curve, where the intersection of the orbit with the vertical line through S may be at any angle whatever, at least within quite a wide angle of range. In some cases this orbit reënters after several revolutions depending on the relation of r_1 and r_0, three revolutions where $r_1 = 10 r_0$, and the maximum range of r in these curves which reënter may be great or small, according to the way the curve or orbit is started, and the tendency is to persist in the way in which it is started; and it may be as in Fig. 234 for one form.

From this it would appear that the governor should be very nearly isochronous, as near as possible without "racing" fits, for securing greatest certainty of regularity of motion.

Mr. Jesse M. Smith.—I am surprised at the position of the curve as shown in Fig. 178, if that curve was taken from a horizontal engine. Why the centre of the inner curve should be displaced horizontally instead of vertically I cannot explain to my satisfaction, unless it be that there is an unbalanced valve-stem or some other horizontal force.

I had experience in this matter in one of the first governors which I designed, where there was a valve-stem about $\frac{3}{4}$ inch in diameter, the pressure in the valve-chest being 90 lbs. to 100 lbs.

That would make a pressure on the valve-stem of about 40 lbs., acting always in the same direction. I found that after setting the valve very carefully while the engine was cold, so that it would cut off exactly the same at both ends, when the engine was started and working under regular conditions, the card at one end would be very much larger than that at the other. I discovered, finally, that it was the effect of the 40 lbs. pressure on the valve-stem acting always in the same direction which shortened the cut-off of one card and lengthened the other, and by running the valve-stem straight through, so that it was balanced, the irregularity in the indicator diagrams disappeared immediately. In the same governor, as shown in Fig. 228 of my own paper, the eccentric *D* is mounted on a shaft, *B*, and has a counter-weight opposed to it so that the effect of gravity on the eccentric is overcome. It will be noticed in this governor, and in all governors which are symmetrical, that the effect of gravity does enter, because while one weight is pulling down the other is pulling up. I was persuaded at one time to cut off the counter-weight of the eccentric, because the governor did not work properly, but after it was cut off the governor worked worse than before. The indicator-cards were distorted. I put the weight back and had better results.

I believe the displacement of the centre of the curve horizontally is due to a horizontal force acting with gravity, and maybe the unbalanced valve-stem.

Prof. J. B. Webb.—I intended to call attention to the fact that the shaft-governor problem is a complicated one when everything is taken into account, and I am glad to see that a number of gentlemen have brought up different views. All of these views are necessarily included in any mathematical treatment. The effect of inertia is well known, as are also the other effects which have been mentioned. We cannot make a complete mathematical treatment without putting all these things in. The special complication in reference to the spring which Mr. Jacobus has mentionèd is this, that the ordinary spiral spring has for its law of extension, when not under the influence of centrifugal force, the simple approximate law that the number of pounds required to stretch it varies directly as the amount of extension. This is not a perfectly correct law on account of the fact that there is some bending of the wire of the spring, whereas the law supposes nothing but torsion. When, also, the spring is put in a fly-wheel, so that its mass is under the influence of centrifugal force, there ensues another variation from the

law. These differences, though small, may be appreciable in some cases where very exact adjustments are necessary. The complete problem of the effect of the centrifugal force of the mass of a spiral spring upon the form of the spring and upon its length under a given stretching force is a complicated one. A simple case has been worked out, under my direction, by Messrs. Willis and Wortendyke in their graduation thesis, in which a spiral spring lies in the plane of the wheel, without coinciding necessarily with a radius, and is supposed to remain straight. In the general case the two points of attachment of the spring are not necessarily in the same plane perpendicular to the axis, and the spring assumes a curve between these points. It is only when the spring is short and stiff that it may be supposed to remain straight. I have analyzed the case of an *inextensible* chain hanging in a fly-wheel; the curve may properly be called a *centrifugal catenary* and involves elliptic functions. One special form of the curve is a right-line, another the arc of a circle, and a third the form assumed by a girl's skipping-rope.

This is a very interesting paper of Prof. Sweet's discussing the simple case of a ball balanced by a string. You will find the same supposition made in a paper which I shall present at this meeting on the length of an indicator-card. A mass oscillated by a spring has a very close connection with a mass under the influence of centrifugal force.

As to the curve deduced by Prof. Sweet, I believe he has neglected to consider one important thing. Gravity acts throughout the two lower quadrants, as he says, in a direction away from the centre, but that does not prove that the weight will be moving away from the centre in both of those quadrants. Gravity acts always in the downward direction on an ordinary pendulum, but half the time the pendulum is rising against gravity. Now, consider a mass sliding radially in a fly-wheel whose centrifugal force is exactly balanced in all positions by a spring. Starting at the highest point, gravity will act throughout the first quadrant to decrease its radius, and at the horizontal position, between the first and second quadrants, the mass will have acquired a velocity toward the centre. In the second quadrant gravity acts away from the centre and gradually reduces this velocity, so that at the bottom point the mass reaches its nearest position to the centre. Then, in the third quadrant, while gravity is still acting away from the centre, the mass acquires a velocity in the same direction which

reaches a maximum at the end of that quadrant, and there diminishes throughout the fourth quadrant, at the end of which the mass reaches its position farthest from the centre. The curve is therefore, neglecting such friction as tends to check the motion of the mass, symmetrical about a vertical line, and has its highest point at the greatest distance from the centre.

One word with reference to the question asked as to Mr. Smith's drawing on the blackboard. I think it is perfectly clear that the effect of inertia does act on the line *ab* as a lever arm, and asked the question to make sure that I understood correctly the statement made.

I have worked on this problem because from a mathematical standpoint it is a beautiful one.

With reference to the effect of the friction of the engine-valve, a change in the amount of this friction will cause a change in the position of the governor. All the work necessary to operate the valve has to go through the eccentric, and if the amount of this work is changed the reaction upon the eccentric will be different, so that the governor must seek a new position to be in equilibrium. When the throw of the valve is changed there is a corresponding change in the work needed to operate it. The inertia of the valve is a powerful element, but its effect is reversing all the time and may be controlled by a dash-pot.

Mr. C. W. Barnaby.—I think the effect upon the weighted arm would also depend largely upon the position of the pivot therein. If the weight arm extended beyond the pivot so that there was a considerable part of the arm on the opposite side of the pivot from the weight, it would have a greater tendency to revolve in opposition to the leverage indicated by *ab* than when the arm was all on the weight side of the pivot, but still the tendency would be there and would modify the effect indicated by the leverage *ab*.

Prof. Webb.—The effect alluded to by Mr. Barnaby exists, and depends on the proportions and position of the ball and its arm being, however, quite small in ordinary cases. If the ball and arm were pivoted at their centre of percussion, with reference to the shaft, Mr. Barnaby's effect would neutralize Mr. Smith's; but in ordinary cases the pivot is far outside that centre.

Prof. John E. Sweet.—Replying to Profs. Robinson and Webb, it is only too evident that Mr. Armstrong and myself were half in error as regards the theory, and this leaves us no ground to stand upon except the fact that the unbalanced eccentric does act in such

a way as to disturb the valve motion of a horizontal engine. How this can be so when theory says the unbalanced weight only tends to throw the eccentric up, I can only reason out in this way:

As we have shown, the action of gravity tends to throw the eccentric directly to one side. The momentum it acquires in one quarter is not overcome until the wheel has made another quarter turn, and hence the limit of the movement is reached when the centre is at the top; but, as Prof. Robinson has proven, a little friction tends to throw the centre to the left. So we can well and safely reason that if we have just enough friction (as may be the case) to neutralize the momentum, then the disturbance will be directly toward one side and make the theory and fact harmonize. That even is of very little consequence compared to the knowledge of the fact that we can now understand why the unbalanced weight disturbs the shifting eccentric-valve motion of the engine.

[NOTE.—This paper received discussion jointly with a paper entitled "A Governor for Steam-Engines" (No. 409), by Mr. Jesse M. Smith, and another on "A Use for Inertia in Shaft Governors" (No. 408), by Mr. E. J. Armstrong. There are references, therefore, in the debate printed with each paper to some of the points brought out by the others.—*Secretary*.]

CCCCVIII.

A USE FOR INERTIA IN SHAFT GOVERNORS.

BY E. J. ARMSTRONG, SYRACUSE, N. Y.

(Junior Member of the Society.)

In presenting this paper to the Society, the writer does not presume the ideas to be new to all. Talleyrand said: "There is nothing new, except that which is forgotten;" and in the exhaustive study which has been given to the many obscure problems of engine governing, it seems improbable that this one should have been overlooked. It has not, however, been generally recognized and the writer has failed to find any literature bearing on the subject.

It seems to be demonstrable that inertia, usually a disturbing element, can be and is made the reverse, even to performing the functions of a dash-pot. To one having some experience with shaft governors, it would at first appear singular that certain governors should be capable of close regulation, without introducing an oscillatory action, fatal to good governing. With a fly-weight moving a considerable distance, a moderate speed, and friction reduced to insignificance, it would seem that were any efforts made toward close regulation, all the conditions for a first-class revolving trip-hammer would be present; and yet these governors run smoothly, even when adjusted very nearly to isochronism, within one-half of one per cent. in some cases.

The reason for this steadiness under apparently adverse circumstances seems to be about as follows: When any governor is engaged in its task of controlling the engine, the fly-weight travels at a variable speed, resulting from its revolving in a circle of variable size; this change in velocity is often quite considerable, depending, of course, on the amount of radial movement and the rotative speed: to take an example from the straight-line engine —a fly-weight is $16\frac{7}{8}$ inches from the centre of the shaft when "in," and $20\frac{1}{4}$ inches when "out," making, at 220 revolutions per minute, a difference in velocity of 194′ 4″ per minute. Whenever

the fly-weight takes a new position it must change its speed—must either move faster or slower, be accelerated or retarded, and either absorb or give out power—somewhere. This resistance to a change in velocity acts at right angles to a radial line drawn through the centre of gravity of the weight, and if the weight were pivoted so as to move radially, as in Fig. 222, the only result would be to put a pressure on the pivot. If the fly-weight is so pivoted as to move at an angle to a radial line, as in Fig. 223, so that in its outward movement it goes toward the way the wheel

FIG. 222.

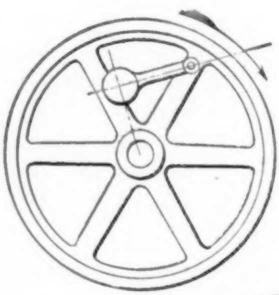


FIG. 223.

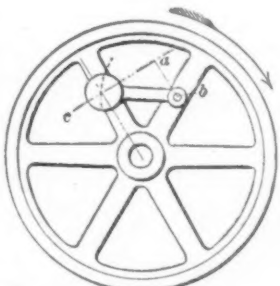


FIG. 224.

rotates, then the outward movement of the fly-weight will be opposed, to some extent, by this resistance to acceleration, depending on the angle which the line of movement forms to a radial line—or, to put it another way—upon the length of the lever arm *ab*, *ca* being the line of resistance, and *b* the fly-weight pivot. When the weight moves toward the shaft, the action is the same. It has to part with some of its momentum, and so hangs back, as in its outward movement, thus making, similarly to a dash-pot, a resistance to movement in both directions, which can only be overcome quickly by a great force, or by a small one

slowly; this resistance increases as the velocity of the weight, which is all which is accomplished by the ordinary dash-pot. Fig. 221 shows a fly-weight pivoted so as to move at an angle to a radial line, but in the opposite direction to Fig. 223. A number of governors have been constructed on this plan, with the idea that any sudden jumping ahead would tend to leave the fly-weight behind, throwing it outwardly and shutting off steam, or *vice versa*, in case of a sudden retardation of the engine; the whole scheme being to make the governor very sensitive to changes in speed. In the variable velocity of the fly-weight we may find the reason why these attempts have not proved more successful. As the fly-weight starts to go "out" it has the velocity due to its distance from the shaft; being pivoted so as to swing back freely, there is nothing to give it the greater velocity due to its increasing distance from the shaft until it gets it all at once at the end of its movement, with a slam not at all reassuring to the designer. Then, when it starts to go toward the shaft it has a high velocity, and does not part with it until it strikes bottom. Of course a dash-pot will cure it; but is it good engineering to build governors that way? At the beginning of this paper, mention was made of the absence of friction as an unfavorable condition. Of course any considerable friction in a governor becomes a serious fault, but a small amount, such as is present in nearly all governors under the most favorable circumstances, would seem to be rather of advantage; the friction and inertia of the valve and its connections are usually sufficient to keep up a slight vibratory movement in the governor, and when this is the case the fly-weight will move very easily if it only moves slowly, the friction coming into full effect should the weight attempt a more rapid change of base. This of course does not apply in the same degree to those governors having a locking or wedging action, and uninfluenced by valve friction or inertia.

DISCUSSION.

Mr. Frank H. Ball.—I wish to congratulate the author of this paper on his clear presentation of a subject which I think has been generally overlooked by engine designers. In the main I accept his theory and conclusions, but not having giving us *all* the theory, some of his conclusions seem to me to be wrong. Referring to his illustrations, Figs. 222, 223, and 224, the effect of inertia felt along

the line AC, or at right angles to the radius, is produced by one of two causes, or by their joint action. These causes we will analyze and will designate them as Cause First and Cause Second.

CAUSE FIRST.—When a change in the *rate of rotation* takes place without a radial movement of the weight, the effect of inertia is felt on the line AC (Fig. 223).

CAUSE SECOND.—When a *radial movement* of the weight takes place with or without a change of the rate of rotation, the effect of inertia is felt on AC, or a line at right angles to the radius.

The theory set forth in this paper is quite correct as to the effects of inertia due to Cause Second, but let us see how it applies to Cause First.

First let it be understood that inasmuch as centrifugal governors depend solely on a slight change of speed for their power to act we always have Cause First to deal with first.

To illustrate: suppose the engine referred to as running at 220 revolutions per minute be suddenly relieved of its load. The consequent acceleration of speed will increase the centrifugal force of the governor weights, which in turn will move outward and reduce the steam supply. Let us suppose at 221 revolutions the increased centrifugal force is sufficient to overcome frictional resistance. But with the arrangement shown in Fig. 223, the inertia of weight due to accelerating rate of rotation, or Cause First, acting toward C, also opposes the outward movement of weight. Therefore the rate of revolution must still further increase until an additional amount of centrifugal force shall overcome both these obstacles to radial movement. Suppose an additional increase of speed of one revolution per minute will suffice to overcome the inertia due to Cause First acting toward C. We then have the following:

Normal speed..	220
Accelerated speed necessary to overcome friction..................	221
Further acceleration necessary to overcome the inertia of acceleration produced by Cause First..................................	222

Applying the same reasoning to Fig. 224, we see that the inertia of weight due to Cause First acts in the direction of centrifugal force, and jointly, they must produce radial motion with a much smaller change of speed than with the other arrangement. So far, then, the better performance is with Fig. 224. The instant radial motion begins the inertia due to Cause Second is felt, and then, as Mr. Armstrong has clearly shown, the better performance is with

Fig. 223, because of the possibility of an adjustment more nearly isochronous without instability. The choice between the two arrangements is therefore between the more prompt action of Fig. 224, and the more perfect or isochronous final action of Fig. 223.

The effect of inertia due to Cause Second very closely resembles the effect of a dash-pot, but the initial effect due to Cause First differs greatly, because inertia not only seeks to retard the radial motion, but it exerts a positive effort to move in the wrong direction, thus preventing the slightest movement outward until the resistance is wholly overcome by increased centrifugal force. In choosing between the two arrangements shown in Fig. 223 and 224, the only point in favor of 223 is the dash-pot effect described by Mr. Armstrong, which is obtained at something of a sacrifice of promptness of action, as has been shown. He admits the desirability of this dash-pot effect, and a fair inference is that he thinks it worth the sacrifice in the way of promptness, and that he thinks this sacrifice less objectionable than the use of a dash-pot. Let us examine this further, first assuming that the dash-pot effect is as satisfactorily obtained with a dash-pot as with the arrangement shown in Fig. 223. Suppose the dash-pot be attached to the weight or weight-arm in Fig. 224. Here, then, we have the dash-pot effect adjustable as to quantity or intensity, and, combined with it, the most favorable conditions for prompt action, as against the arbitrary and unadjustable dash-pot effect in Fig. 223, combined with more unfavorable conditions for promptness of action. Stopping right here, the choice must certainly be with the dash-pot. As against the dash-pot, however, is the additional cost, complication, and care. If the dash-pot be attached rigidly to the moving parts of the governor, as is commonly the case, other questions arise. For instance, to what extent is this force beneficial, or what are its useful limitations? The function of a dash-pot is to offer resistance to rapid motion in either direction. Disregarding friction, this resistance is controlled by the size of orifice in piston, through which the fluid is made to pass, and with openings of all sizes the resistance is unappreciable at very slow rate of motion. The nearer approach to isochronism in the adjustment of the governor, the more the dash-pot is depended on for stability, and the smaller must be the piston orifice. A difficulty is here encountered, for in reducing the piston orifice, the slower motion of piston, if attached directly to the weight or weight-arm, produces a sluggish motion of the weight, and practi-

cally prevents wide and sudden radial motion, which is necessary under sudden changes of load. It is probable, however, that a dash-pot effect equal to that obtained in Fig. 223 could be satisfactorily used, although a much nearer approach to isochronism may not, in practice, give satisfactory results.

Fig. 247 shows another application of the dash-pot which combines other features.

In this case the pivot of weight is relatively the same as in Fig. 224 in respect to the effect of inertia, except that the outward movement of weight decreases the inertia effect by bringing the pivot more nearly at right angles to the radius, which would seem to be better than the opposite condition of Fig. 224, because the outward

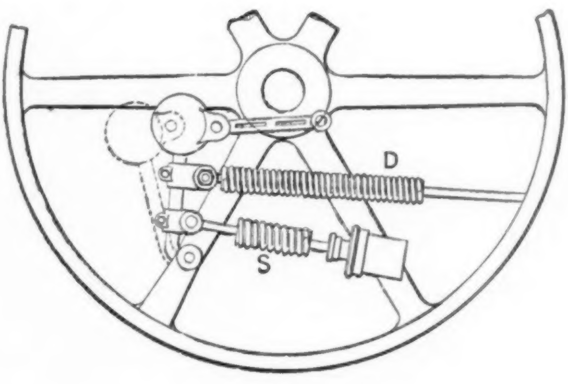


FIG. 247.

movement of the weight is more likely to be too rapid than the inward movement, for reasons which need not here be mentioned. The dash-pot here shown is connected with the weight-arm by means of a spring, arranged for either extension or compression, and having a range of deflection sufficient to allow the extreme motion of governing mechanism without any motion of the piston whatever. The piston orifice is supposed to be sufficiently small to insure a very moderate rate of piston movement. The dash-pot spring is supposed to offer only sufficient resistance to radial motion of the weights to insure stability when the governor is adjusted to perfect isochronism. With this arrangement the effect produced when radial motion occurs is not strictly a dash-pot effect, as the dash-pot need not necessarily act. Until piston movement occurs the

effect is only that produced by deflecting the dash-pot spring, which, having no initial tension, compels stability by making the governor for the moment "non-isochronous." The piston movement, which should occur almost simultaneously, relieves the dash-pot spring of its tension, and leaves the governor in its isochronous condition, simply having prevented instability and still standing guard over radial movements for that purpose only, leaving the speed to be determined by the balanced centrifugal and centripetal forces adjusted to isochronism.

Mr. Ezra Fawcett.—In my experiments with shaft governors,

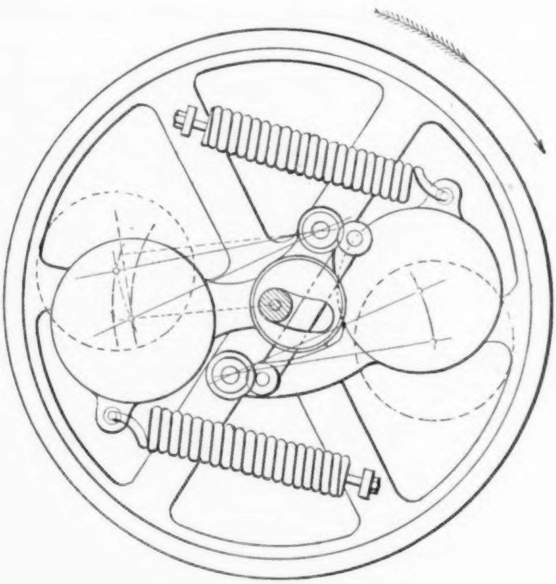


FIG. 237.

and facing the usual trials incident to such work, some time since I set about designing a governor which would obviate some of the difficulties. My plan was to project the fly-weights at a point similar to that shown in Fig. 223, such that they would travel at the speed and velocity which should vary the least (Fig 237). The arm, eccentric, and weight were in one casting, and the attachment of the springs was at the extreme points of the weights. My troubles from this cause ceased, and a variation of less than one per cent. could be secured, and no perceptible variation of speed with variable boiler pressure.

One peculiar point observed was that the speed could not be reduced much less than the normal rate of 400 revolutions per minute, with a governor-case of 17 inches diameter, path of fly-weights 12 and 14 inches in diameter, by adding to or reducing the weight and strength of springs, provided only they were sufficient properly to work the eccentric and valve.

Mr. Jesse M. Smith.—I was very much interested in Mr. Armstrong's paper, and looked into it quite carefully. In Figs. 238, 239, and 240 are shown the three cases given in Figs. 222, 223, 224 by Mr. Armstrong. In all cases the fly-weight has two distinct motions, one of rotation about the centre of the shaft, and one of rotation about the centre of suspension b. Either or both motions may be constant or variable, and they may occur at the same time or different times.

Suppose, first.—No motion about b and a constant motion about shaft O; there is evidently no force of inertia developed.

Second.—No motion about b and a variably increasing motion about O; the force of inertia will then act in the opposite direction to the rotation, and at right angles to a radius passing through the centre of gravity of the weight and the centre O.

Third.—No motion about b and a decreasing motion about O; the inertia will then act at right angles to the radius, but in the same direction as the rotation about O.

In these two last cases the inertia acts just as stated by Mr. Armstrong.

Fourth.—Motion about b outward, and increasing motion about O.

Suppose in Fig. 238 that while the weight moves from c to c' about b it also revolves about O through 45°; the weight will then travel on a curve, cc'', whose form will depend on the kinds of motion about b and O.

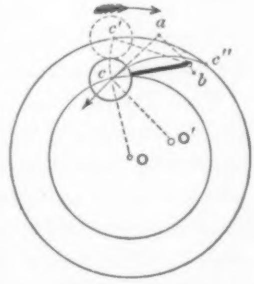


FIG. 238.

Now, a weight travelling with a variable motion on a curved path develops a force of inertia which is tangent to the curve, and a centrifugal force which is normal to the curve at the point occupied by the weight.

Suppose, to simplify the question, the curve cc'' is a circle having its centre at O', and that the motion is increasing; the force of inertia will be in the direction of line ac, and will act backward on the lever arm ab, and not directly on the pivot, as stated by Mr. Armstrong.

The force of inertia will tend to hold the weight back and make it move more slowly. In the case shown in Fig. 239, under the same supposition of an increasing motion on the curve *cc''*, the inertia acts in the same way, but has a longer lever arm, *ab*. In the case shown in Fig. 240, under the same supposition, the force of inertia passes nearly through the centre *b*, and acts on a short lever *ab* to move the weight in the same direction in which it is travelling, but it acts with much less effect than would seem to be indicated by Mr. Armstrong's statement. It will be seen what a complicated question this gets to be if an attempt is made to consider the conditions of practice and how very different the results will be under different conditions.

The rate of rotation about the shaft depends on the weight of fly-

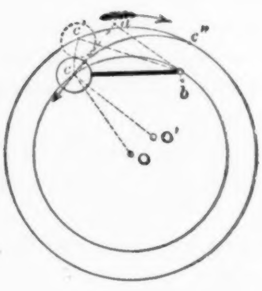


Fig. 239.

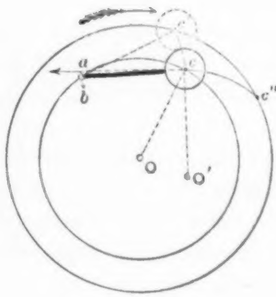


Fig. 240.

wheel, the rate and amount of change of load, and the prompt action of the governor.

The rate of motion about the pivot *b* depends principally on the friction of the governor joints and the adjustment of the springs.

Each one of these causes has its effect, and few of them are constant or controllable. If, however, friction be reduced to a minimum and heavy fly-wheels are used, the forces which cause the governor parts to change position act slowly, and the parts take their new positions promptly, but slowly, so that the forces of inertia are not developed to an objectionable extent.

Mr. C. M. W. Smith.—Mr. Armstrong has represented three cases in his paper. But I believe the most important case is omitted, and with your permission I will make a drawing on the board to represent it. Now, you will notice in Fig. 223 that as the weight moves outward the arm *ab* increases in length, and not

only this, but the inertia of the weight also increases, and so we have two increasing elements, which tend to resist the weight as it moves outward. Now, in Fig. 241, as the weight moves outward the inertia increases, but the arm *ab* decreases. Now, what I am getting at is this, and I wish to make this point, that the arm *ab* should be of such a length—that is, the forces should be in such a relation—that the inertia will overcome the friction of the governor, thus having a governor practically without friction.

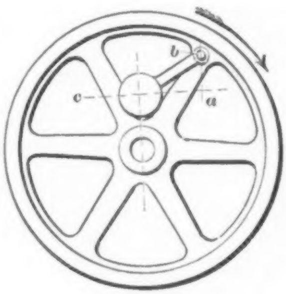


FIG. 241.

He mentions the fact in his paper, that a little friction is a good thing, but we cannot control this amount of friction, as in practice sometimes the governor does what is called sticking, caused by bad lubrication and neglect, and the engine must increase many revolutions in order to overcome this sticking, and when the weight does start it goes to its outer position with a slam. Now, by arranging the arms in this relation (Fig. 241) the friction can be overcome, because the inertia can be so proportioned that it overcomes the diminishing in length of the lever arm *ab*.

There is a governor in existence where the centrifugal force of the weight is exactly opposed by the pull of the spring. A dash-pot and a secondary spring are attached to resist the motion, so that isochronism can exist. What I have said is what I have learned from experience with different classes of governors.

Mr. Jesse M. Smith.—I would ask Mr. C. M. W. Smith in what direction—taking his figure on the board—in what direction the force of inertia acts, or in what direction he considers that it acts?

Mr. C. M. W. Smith—As the arrow indicates, the wheel is moving in that direction. The inertia holds the weight back—that is, it must overcome the increase in speed.

It takes power to increase the speed of the weight as it flies off from the centre, and this pulls back on the weight.

Mr. Jesse M. Smith.—I do not think that Mr. Smith quite gets my idea. I would like to know in what direction the force of inertia acts, whether it acts at right angles to a line drawn through the centre of the weight and centre of rotation, or in a different line?

Mr. C. W. Barnaby.—As I see it now, I think there would be two tendencies to take into consideration in considering the action of the weight arm under the conditions Mr. Smith refers to. In the first place, if the rotation of the governor-wheel is retarded, the governor-ball, revolving within the pivot *a* (Fig. 231), might, as Mr. Smith has stated, have a tendency to move ahead toward *b*, acting on the leverage *ab* to rotate the weighted arm on its pivot *a*; but there is another effect to be taken into consideration, I think, and that is the rotation of the weighted arm about its own centre of gravity. Of course this arm in making one revolution around the centre of rotation of the wheel makes one revolution upon its own centre of gravity. If the rotation of the wheel is checked, the weighted arm itself would have a tendency to continue revolving in the same direction with unchanged velocity, upon its centre of gravity, which would tend to move the weight from *b*, or in the opposite direction to the first tendency considered.

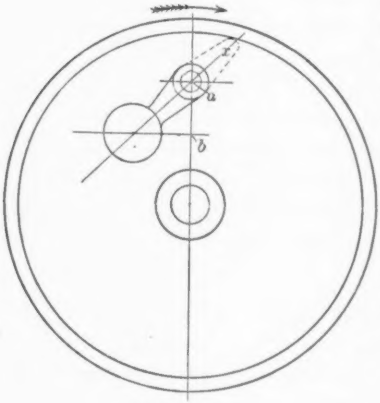


FIG. 231.

The actual effect upon the weighted arm would be the resultant force due to these two effects. There is, I think, a certain circle within the pivot *a* at which these two effects upon the weighted arm would just balance each other, but when the weight was revolving inside of that circle, the result of any check in the speed of the governor-wheel would be to throw the weighted arm still further in, and when revolving outside of that circle the weighted arm would be thrown further out, although the leverage *ab*, as indicated by Mr. Smith, might show the opposite result at this point.

Prof. J. B. Webb.—Are we to understand that the previous speaker said that the increase of speed would prevent the ball going out—that is, that the effect of inertia and the increase of speed would be to hold the ball in?

Mr. C. M. W. Smith.—I did not mean to say that, because it acts on the arm *ab* and would tend to throw the ball out and overcome the friction of the governor.

Mr. Jesse M. Smith.—The question is, Does it act on the arm *ab?*

Mr. C. S. Dutton.—I would like to ask the speaker if we understand him to say that a governor is in existence which does maintain an absolute speed? I understood him to say something to that effect. I am reminded a good deal of what a friend of mine, a member of this Society, said in regard to a telephone. He was building a little telephone for use in his own house and experimenting with it, and during that time—it was several years ago, in the early history of the telephone—being in Cleveland, he went into the Electrical Supply Co.'s place to get some supplies of wire and one thing and another, and meeting Mr. Brush had some talk with him on various matters, and among other things mentioned to him that he had made a little telephone in his house, and that to his surprise it worked very nicely. Mr. Brush said: "Well, now, if you told me that you had made a telephone and it did *not* work I would have been very much surprised."

Mr. Jesse M. Smith.—I did not wish to have Mr. Dutton or any of the gentlemen here present think that there was a governor in existence which would prevent some change in speed. I am painfully aware of the fact myself, and I think that any one who has designed a shaft-governor and brought it out, so that it is commercially satisfactory, has been painfully aware of the many misfortunes which a governor must meet before it arrives at that point.

Mr. E. J. Armstrong.—The point raised by Mr. Ball is rather hard to reply to from a purely theoretical standpoint; the effect is, as he has pointed out, to make the governor tardy, but it is, I believe, always found best to have a much slower movement of the fly-weight than theory, as usually elaborated, would indicate; and as the governor under discussion attends to its business about as closely as any, possibly the tardiness referred to is quite within the limits of good governing. If a governor will, as this one does, move from one extreme to the other in less than a second, when the whole load is instantly thrown off or on, it is probably about as quick as it can be and retain its stability. In those examples of the dash-pot governor described by Mr. Ball which I have had the pleasure of examining, the auxiliary spring connecting the fly-weight and dash-pot has been so large and stiff that it is hard to see any essential difference between it and a rigid connection, which goes to show that some tardiness is not objected to in that governor, for it must take some force to bend that stiff spring. With regard to Mr.

Jesse M. Smith's statement that the inertia acts tangent to the curve described by the fly-weight in its outward or inward movement, and not at right angles to its radius, this curved path is the result of two motions—one around the wheel, and the other in or out from its centre. This inertia effect is caused by the changing velocity of the weight caused by a radial movement. The velocity changes in a direction not tangent to the curve the weight may be describing, but at right angles to its radius; consequently inertia will act on this line, and not tangent to the curve. The point may be clearer if we conceive the radial velocity to be very high, so that the curve becomes almost a radial line. Surely no one would contend that the inertia effect would be tangent to such a line.

Mr. C. M. W. Smith has shown (Fig. 241) an arrangement of parts similar to Fig. 224. The direction in which the leverage increases is of little moment, for the fly-weight has to go in as well as out; and, under the same conditions, I must disagree with his statement that "the forces should be in such a relation that the inertia will overcome the friction of the governor, thus having a governor practically without friction." It is impossible to balance friction, a presumably constant quantity, by the inertia of the fly-weight, which changes with its velocity. Then, too, this inertia force does not exist until the weight has moved, and the force he would have overcome friction is not generated until after the friction has been overcome. I would suggest that a governor built on that plan would require an exceedingly business-like dash-pot to control it. Mr. Smith has also made the evident error of presuming this inertia force to be greater as the fly-weight goes further from the centre of the shaft; a little consideration will show that radial distance has no effect. The radial velocity and the rate of rotation are the only factors; these being constant, the acceleration of the fly-weight will be the same whether its radius be an inch or ten feet. After all, fly-wheel governors are hard things to theorize about—there are so many unknown quantities among the factors. That the governor with the fewest parts and most severe mechanical simplicity should do what others require so much complication to perform, would seem to be evidence of its correct design.

[NOTE.—This paper received discussion jointly with a paper on the effect of an unbalanced eccentric, by Prof. J. E. Sweet (No. 407), and another by Mr. Jesse M. Smith, entitled "A Governor for Steam-Engines" (No. 409). There are, therefore, references in the debate printed with each paper to some of the points brought out by the others.—*Secretary*.]

CCCCIX.

A GOVERNOR FOR STEAM-ENGINES.

BY JESSE M. SMITH, DETROIT, MICH.
(Member of the Society.)

THE governor, which is the subject of this paper, was designed on the basis of the following propositions:

First.—A governor to be sensitive must be as free as possible of friction.

Second.—To be powerful the forces which are in equilibrium must be large compared to the resistance of the valve to be moved.

Third.—In order that the shaft may not be thrown out of balance by change of position of the governor weights, these weights must be symmetrical.

Fourth.—That the engine may make long runs the joints of the governor must be so constructed as not to require oil, or be capable of lubrication while in motion.

The centrifugal governor has assumed an almost infinite variety of forms, from the fly-balls of Watt to the shaft governor of the modern high-speed engine. In all forms, however, the motive power is the centrifugal force of a weight. The centripetal force, which equalizes the centrifugal force of this weight, has been supplied by gravity, by the pull of the main belt, by springs, by inertia rings, and by the centrifugal force of other weights.

Various devices have been added to prevent or modify the "racing" of governors due to too sensitive adjustment or friction in the joints. Air and fluid dash-pots, connected directly and through springs to the moving parts of the governor, have long been used with more or less benefit.

The lamented John C. Hoadley was among the earliest to design a shaft governor in which the valve was actuated as well as regulated by the governor. He has had many followers who have profited by his experience and added their mite, until we have to-day many governors which give excellent results. In every governor the forces are in equilibrium, just as the weights

on a scale beam are in balance; and whenever this equilibrium is disturbed by change of speed the parts seek new positions in which they will be again in equilibrium. The ease and therefore the promptness with which the parts move to a new position depends, not upon the resistance of the valve, nor the friction of the parts which are constantly in motion relative to each other, nor to the weights, nor springs, but to the friction of the parts which were at rest relative to each other. That is, the friction of the joints of the governor proper, beginning at the eccentric disk and ending with the flying weights.

The friction of the valve, valve connections and eccentric strap, taken as a whole, is a variable quantity, passing from zero to a maximum and back to zero twice in each revolution. Its mean value, however, is, or may be made, reasonably constant by the modes of lubrication in general use.

The friction of the joints of the governor is quite another force. With a constant load and steam pressure the parts assume a certain position, and do not leave that position until after a change of load or steam pressure takes place. The parts being in relative rest, they adhere to each other somewhat, and if the oil becomes gummy or scarce, as it may during a long run, they adhere quite firmly. When the change comes, the forces try to overcome this *friction of rest*, but the instant motion begins the friction is greatly reduced, and the forces which overcame the friction are unbalanced, and the weights move way past their proper position. In the effort to overcome the friction the speed has changed from the normal, and when the weights do move they move too far and allow the speed to change in the opposite direction. We then have "racing" more or less severe, according as the joints are well or poorly lubricated. It is this friction of the joints of the governor, and not the friction of the valve and connections, which prevents the use of the isochronous governor, or an adjustment of spring closely approaching isochronism.

In a successful governor for close regulation it is not only necessary that friction of the joints be small when the engine is first started, but that it will remain small during a long run of, say, 200 or 300 hours. It is therefore very desirable that the joints be made so as to require little oil, or be capable of lubrication, positively, while in motion.

The governor, which is the subject of this paper, was designed in 1883, and has been applied to a number of engines with unbal-

anced as well as balanced valves. It is shown in Figs. 228 and 229.

A small shaft *B* is journaled in the hub of the fly-wheel, and is parallel to the main shaft. The eccentric, whose centre is at *D*, is fixed to one end of the shaft *B*, and the cross-arm *d* to the other. The centre of the eccentric may thus move about *B*,

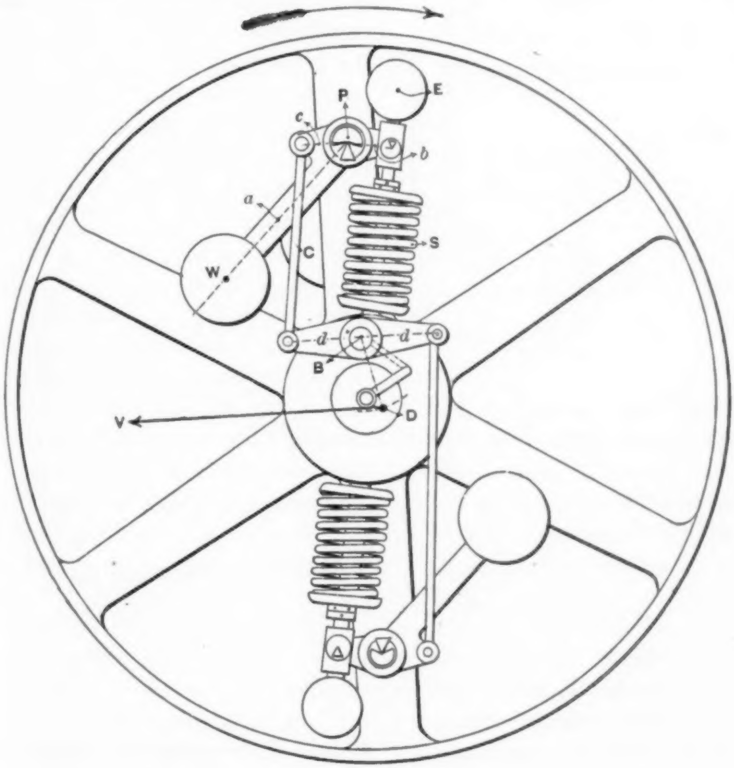


FIG. 228.

across the shaft, and produce the variable valve motion. Each end of the cross-arm *d* is connected by a link *C* to an arm *c*, pivoted at *P*. The flying weight *W* fixed to the arm *a*, also pivoted at *P*, tends to move outward as the speed increases. It is resisted by a weight *E* acting on the arm *b*, also pivoted at *P*, which moves inward when *W* moves outward. The spring *S*, whose axis is radial, also acts on arm *b*, and assists the weight *E* to urge *W* inward. The valve resistance *V* also assists the weight *E*. The arms *a b c* are all formed in one piece. The

weights W and E and spring S move as nearly as possible upon radii from the centre of rotation. For the purpose of reducing the friction to a minimum, the pivot P, which sustains the greatest strain, and the bearings at the ends of arms b, are made in the form of knife edges of hardened steel. They require little or no oil, and are enclosed so as not to gather dust. The joints of the links C support little strain, and are usually made simple pin connections. The eccentric being mounted on the small shaft B, which has a long bearing in the hub of the fly-wheel, requires little force to move it. The shaft B may, besides, be oiled while the engine is running, by means of a small pipe extending from the centre of the main shaft to the middle of shaft B, so that the friction here is also reduced to a very small amount. This governor is readily adapted to run in *either* direction. The forces being in equilibrium, we may write the following equations:

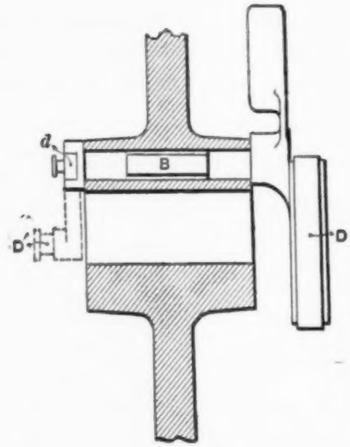


FIG. 229.

$$WRN^2\ Fa = ErN^2\ Fb + Sb + \tfrac{1}{2}\ Vc. \quad \text{Equation (1).}$$

$$WR'N'^2Fa = Er'N'^2Fb + S'b + \tfrac{1}{2}\ Vc. \quad \text{Equation (2).}$$

In which

N = Maximum revolutions per minute.

N' = Minimum " " "

R = Distance from centre of weight W to centre of rotation at N rev.

R' = Same at N' rev.

r = Distance from centre of weight E to centre of rotation at N rev.

r' = Same at N' rev.

S = Tension of spring at N rev.

S' = " " " " N' rev.

V = Resistance of valve and its parts as applied to centre of eccentric D.

F = Centrifugal force of 1 lb. 1 inch from centre at 1 rev.

The first side of the equations represents the moment of the centrifugal force of W about pivot P. The second side represents the moment of the centripetal forces about the same point. Suppose, now, the parts to be so chosen that the centrifugal force of the weight E, due to increase of speed, shall be just equal to the decrease of centrifugal force, due to the weight E moving nearer to the centre; then

$$ErN^2F = Er'N'^2F$$

$$rN^2 = r'N'^2$$

$$r = \frac{N'^2}{N_1^2}\ r'$$

Subtract equation (2) from equation (1):

$WRN^2Fa - WR'N'^2Fa = (S - S')\ b.$ Equation (3).

The weight E may evidently be so chosen that in equation (2) the tension of the spring may be zero. That is, the initial tension $S' = 0$.

From equation (3) we then get:

$$S = \frac{a}{b}\ (WRN^2F - WR'N'^2F),$$

which means that the spring has only to take up *the difference* of centrifugal force of the weight W in its inner and outer positions, instead of the *whole* of that force, as in most governors. The spring may therefore be small and short, and still not be strained to such an extent as to fatigue the metal. A common compression spring, such as is in use under cars, has been used, and it is found simple and effective. If a spring breaks, the engine stops. The initial tension of the spring, which is what supplies the greater part of the centripetal force in most governors, is here replaced by the centrifugal force of the weight E, which force is practically constant within the range of speed variation. It may be urged that the inertia of the weight E will retard the action of the governor, and when it is in motion carry the weight too far, and thus be a detriment. On the other hand, the inertia of E and W tends to prevent the change of position of the parts, due to the rapid and great changes of the valve resistance.

The mean value of this valve resistance is practically constant, while the centrifugal forces of W and E change as the square of the revolutions; so that a pound or two increase of these weights

increases enormously the strength of the governor to overcome the valve resistance. Heavy weights are therefore an advantage instead of a detriment, as they prevent the valve from controlling the governor by their inertia and produce a governor strong enough to overcome any valve resistance. Results of practice show some of the best regulations with the initial tension of the spring zero, which corresponds to the greatest value of weight E. It is evident that equation (2) may be satisfied with any values of S' and E

$$\text{from } S' = 0 \text{ and } E = \text{maximum}$$
$$\text{to } S' = \text{maximum and } E = 0\,;$$

but the smaller E becomes, the larger and longer the spring must be; and it has been found in practice that a sensitive, powerful governor can best be had by giving E a considerable value.

In every centrifugal governor a change of speed is necessary to produce a change of the position of the parts, and of the valve motion. Whether this change of speed be momentary or continued, great or small, it is necessary; and the problem to be solved is how to reduce the change of speed to a minimum and still have a stable governor. In the governor under consideration, the regulation is obtained by heavy weights and light springs arranged in a peculiar manner, with friction reduced to a minimum, in contradistinction to light weights and heavy springs with artificial friction introduced to mollify the evils of real friction. As applied to a $9\frac{1}{2} \times 12$ engine, with piston-valve, some of the data are as follows :—suppose

Maximum revolutions	$N = 303$
Minimum revolutions	$N' = 300$
Variation of revolutions	1%
Valve resistance varies from 0 to 20 lbs., mean value	$V = 10$
Initial tension of spring	$S = 0$
Moment of centrifugal force of W at N revolution	$= 2930$
Same at N' revolution	$= 1850$
Increase of moment of W for 1% variation of speed	$= 1080$
Percentage of increase	$= 58\%$
Moment of valve resistance	$= 12.5$
Ratio of valve resistance to increase of moment W	$\frac{12.5}{1080} = 0.0116 = 1.16\%$
Maximum tension of spring	$S = 470$ lbs.
Compression of spring	$= 1$ inch
Moment of centrifugal force of E at N revolution	$= 1744$

Same at N' revolution	=	1831
Difference of moments of E	=	93

If parts had been so chosen as to make this difference = O, the maximum spring tension would have been $S = 432$ lbs. Note that the moment of the valve resistance is very small compared to the moment of the centrifugal force of the flying weight—that is, about 0.4%, or 1.16% of the difference of this moment of the flying weight for a variation of speed of 1%. The valve motion permits a variation of cut-off of from 0 to 0.7 of the stroke. A variation of speed of less than 1% between no load and 0.7 cut-off may be readily obtained in practice, and this regulation can be maintained during long-continued runs. When this governor is applied to centre-crank engines with valve connections out side of the fly-wheel, as is becoming quite popular, the eccentric can be dispensed with, and replaced by a wrist-pin D' formed on the end of an arm extending from the cross-arm d, as shown in dotted lines to the left of the lower figure. This will do away with the annoyance generally attending the use of eccentrics

DISCUSSION.

Prof. John E. Sweet.—The author of the paper, in his third proposition, says: "In order that the shaft may not be thrown out of balance by change of position of the governor weights, these weights must be symmetrical."

It so happens that this statement is not true. In a single-ball governor, if the weight of the eccentric and its attachments balance the governor-ball, and moves in the opposite direction on a line through the centre of shaft, it will balance it in all positions. Even were this not true, in the single-cylinder engine the unbalanced weight of the reciprocating parts exceeds the weight of the governor ball as, perhaps, ten to one, so that the unbalanced single ball at its worst would be unnoticeable in the engine. If the weights shown at A and B, Fig. 243, be coupled by the lazy tongs arrangement, it goes without saying that they will balance in all positions, and that the centre of gravity of the system is in the centre of the shaft.

In Fig. 244 the same system is moved to one side of the shaft, and the centre of gravity is at C, no matter to what extent the balls A and B are moved.

Then, if a counter weight be placed in the wheel at D, the

wheel will not be thrown out of balance by any shifting of the governing weight *A*, and that holds good even if the weights *A* and *B* do not move on a radial line, provided they move in opposite directions. As direct evidence, we run the single-ball governed engine at all speeds while resting on a greased iron plate without any fastenings whatever, and without any evidence that the single ball disturbs the balance.

I am unable to understand, and hence unable to refute, Mr. Smith's mathematics, but in one at least of his conclusions I am confident he is in error. If the weight shown at *E*, in his drawing, has sufficient centrifugal force to resist the centrifugal force of the weight *W* at 25 revolutions, it will do so at 250, or any

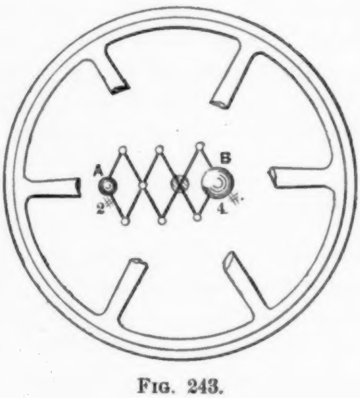


FIG. 243.

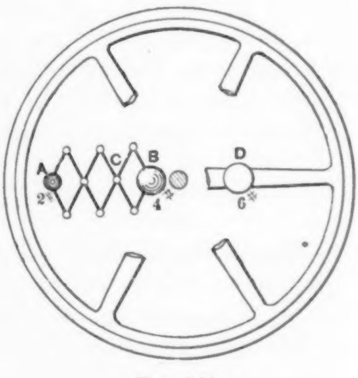


FIG. 244.

other number of revolutions. So that it hardly seems true that the governor can be made to work without more or less initial tension on the springs, nor does it seem as if there would be any great gain in doing so, if it could.

Were all the balls concentrated into one, its inertia would be equally efficacious in resisting the disturbing influence due to friction of valve, and as the one weight would have a greater range of motion than the average of the four, likely it would be more serviceable than in the form shown. The best part of the paper is the statement that "the governor works well," which I am ready to believe (if the weight *E* is small enough), as I am that there are others which work well also, with fewer pieces.

Mr. F. E. Idell.—In August, 1888, I tested three engines having governors such as described by Mr. Smith. These engines are

in the power station of the Sea Shore Electric Railway Co., Asbury Park, N. J.

The station is equipped with four $10\frac{1}{2}'' \times 12''$ engines, each belted to a line shaft in the centre of the room, which could be separated into two parts by throwing out a friction-clutch coupling.

The engines were all set to run at 300 revolutions per minute under 80 lbs. steam pressure. The ratio of the pulleys on the engines and the line shaft was as 8 is to 9. The normal speed of the shaft was, therefore, 337.5 revolutions per minute.

Before commencing the test the four engines were run in pairs to see if they drove the shaft at the same speed. The friction coupling was thrown out so that each pair of engines would drive an independent shaft. A tachometer was placed on the floor at one end of the line and belted to the shaft, and the speed noted. Then the tachometer was taken to the end of the second length of shafting, belted to it, and the speed noted again. The difference in the readings was one revolution per minute, or, at the speed the shafting was running—337.5 revolutions per minute—the error was about $\frac{3}{10}$ of 1%.

During the test, readings of the tachometer were taken half hourly. The average of 61 readings for the two days was 335.5 revolutions per minute, or less than $\frac{7}{10}$ of 1% below the normal. The readings of the tachometer were taken at the stated intervals without regard as to whether or not there was a sudden demand for excessive power at that time. The demand for power was fluctuating constantly. I watched the tachometer for minutes at a time and noticed that the variation in speed lasted but a few seconds, recovering the normal speed within 15 seconds from the time any sudden demand for power was made. Two indicator cards were taken from each engine every 20 minutes during the day and half hourly during the evening. The speed of the engines was taken for half a minute after each set of cards, and the record was so uniformly so near 300 revolutions per minute that this figure was used in working up the cards.

Mr. E. J. Armstrong.—The first proposition of this paper is, "A governor to be sensitive must be free from friction." It is, perhaps, an act of temerity to question this statement, which has almost passed into an axiom among engine-builders, but as the writer has in another paper made mention of good effects from a small amount of governor friction, a statement of reasons for that belief may be in order. In several experiments made by the writer some time

ago to find out how much better governing might be obtained by the use of antifriction devices, the result was not at all as was expected. The governors became less steady, and in order to work as satisfactorily as before had to have the initial tension of spring reduced so as to govern less closely; apparently showing that the small amount of friction originally present was an element of stability and acted to prevent too rapid, and consequently too great, changes in the position of the fly-weights. The valve resistance changes in direction so rapidly that in a reasonably heavy governor the inertia is sufficient to prevent any serious disturbance of the governor system by it. But, unless the governor friction is greater than the valve resistance, it would seem that there must be some vibratory movement caused. If a change in the engine speed occurs, acting to change the position of the fly-weights, the amount is aided half the time by the valve resistance and half the time opposed by it; the sum of the two forces will move the fly-weights much further than their difference will, consequently the fly-weight will hitch along until the changed position of the valve again permits a balance of the centrifugal and centripetal forces. This is offered rather as a theory than as a statement, the experiments referred to not having been carried far enough to prove that this action actually takes place, only to suggest it as a probable explanation of the behavior of the governor experimented with, and of the thousands of governors successfully running in which the governor friction is greater than the difference in centrifugal force due to the change in speed when the load is varied. In the paper under discussion mention is made of the centrifugal force of the weight acting in lieu of the initial tension of a spring. Now the use of a spring in a fly-wheel governor is to furnish a resistance constant for any given position in its ratio to the centrifugal force of the fly-weight, which varies with every change of speed, in order that when a change of speed comes the difference between the constant and inconstant forces may act to change the position of the governor system. On the sixth page of Mr. Smith's paper he says: "Results of practice show some of the best regulations with the initial tension of the spring zero," which means that for one position of the governor the spring has no strain upon it, and that the entire centrifugal force of the fly-weight is opposed by the centrifugal force of another fly-weight. For that particular position of the fly-weight there can be no difference caused by speed between the two forces in equilibrium. If they will balance each other at one speed they

will do so at any other, and it is hard to understand how any governing can be obtained under the conditions; for any other position of the governor the conditions are different. The fly-weight W has moved away from the shaft, and the weight E has been drawn close, so that W has a greater centrifugal value than E, the difference at the speed desired for the governor being supplied by the spring, which is now under tension; when a change of speed occurs the force acting to move the governor is the difference in centrifugal force which would be caused by the change in speed, for a fly-weight having a centrifugal value equal to the difference between W and E, an amount which would seem rather small for quick governing, and liable to make sluggish action of the governor with a considerable less value for E; the force acting to move the fly-weight when a change of speed occurs becomes greater, and the governor becomes of greater power and stability, capable, as it has proved itself, of successful governing under the most exacting conditions. It may put the whole problem in a clearer light to say that for any given position the governor, so far as governing is concerned, is the same as it would be if the fly-weight only had a centrifugal value equal to the difference between W and E, and the remainder of the weights were replaced by one of equal inertia divided between the eccentric and its fly-weight. Only an amount of fly-weight which would balance the spring can be taken as having anything to do with shifting the eccentric, and this amount varies with the position of the fly-weights; the remainder of the weight is simply inertia weight.

Prof. R. H. Thurston.—I am much interested in the paper and the new form of governor which it describes, and would be glad to get more particulars of its performance, especially as to the manner in which it meets those sudden variations of load often met with, especially in electric lighting—those which take place in an instant and without preliminary fluctuations. The principles enunciated in the introduction to the paper are well stated, and it is to meet the requirements thus formulated that inventors of governors have been seeking new and more perfect devices from the days of Sickles' and Corliss' inventions, and especially since the demands of the finer textile manufactures and of electrical work have become so exacting.

I would add one more to the list, and say:

Fifth.—That the governor may act not only promptly and powerfully, but with accuracy, it must include some adjustment or some

special mechanism which shall insure the quick movement of its parts into the required position, to give equilibrium between driving power and load, between effort and resistance, instant by instant, and thus evade the ill effect of inertia in throwing the governor beyond the right point at each sudden change.

There are, I think, two ways in which this result may be attained: The use of a dash-pot, offering rapidly augmented resistance with increasing velocity of displacement, while allowing moderate rates of movement without appreciable impediment, is a familiar method of meeting this demand, one which has been long in use in nearly all classes of construction. In some cases I think it may be found that the regulation of the exaggerating action of inertia is effected by the friction of the valve-motion itself, which pins the mechanism fast at certain points in each revolution, and compels the exact adjustment of the steam-distribution to but a limited extent in each of several revolutions. How far this may prove useful or desirable action it is impossible at present to say. It would be an interesting matter for investigation.

In the form of governing which is here described I see what appears to me a very interesting and probably useful arrangement, which, whether so intended or not, will perhaps prove to give a similar effect. The weight W is so suspended that any quick movement of the engine will jerk the wheel ahead or back, accordingly as the load is dropped off or thrown on, in such a manner as to produce, in consequence of the inertia of the weight, a relative motion of W and the other parts of the governor, which must result in quick closing or quick opening, as the case may be, so as to meet the more promptly the tendency to acceleration by a responsive action of the valve-motion. The substitution of knife-edges for pins and joints of the usual sort is obviously an important modification in the direction of improvement, and especially as insuring a constant and permanent sensitiveness. It is unquestionably very desirable to dispense with all lubricated joints in any governor of this class. It is in these directions that Professor Sweet has been so long and so succesfully working, and with this end in view, we have just attached to the first of the straight-line engines, that built in the Sibley College shops of Cornell University, a governor designed by Mr. E. S. Bowen, of the University, which is without joint, and absolutely frictionless. It has proved to be most satisfactory; the steam distribution and the regulation have been found all that could be desired, and more than would

usually be expected of any governor of any one of the older forms. Nevertheless, where provision is made for steady and sure lubrication, the action of the original Hartwell governor, or of any of its better class of successors, can usually be made more accurate than any of the Watt governors. I should expect thoroughly good work from the governor here described.

Mr. Frank H. Ball.—The special feature of interest in this paper is the peculiar use of a supplemental centrifugal weight, to produce, in effect, centripetal force, and the author's very ingenious application of this force to take the place largely of springs in a governor problem. The theory is a very pretty one, and is sustained by successful practice. It is the purpose of the writer in this discussion to examine the advantages and disadvantages which belong to the peculiar use of the supplemental weight.

The principal advantage seems to be in the simplification of the spring problem and the possibility of a governor with powerful opposing force in equilibrium without the use of heavy and long springs. For the sake of comparison, let us imagine a governor similar to that shown in Fig. 228, with the same weight W, but without the supplemental weight E, and let us suppose the spring S be increased in strength and length until it shall furnish all the centripetal force required to oppose the weight W. Let us imagine that the spring has the same margin of reserve strength which the author describes as necessary to prevent fatigue of the metal, and also that the relative adjustment of centrifugal and centripetal forces be made to harmonize exactly with the forces in Fig. 228. Obviously, then, both arrangements would seem to show corresponding rates of speed in controlling an engine under similar conditions of load or boiler pressure, and both would be equally powerful. The advantage of cost would be with Fig. 228, having small and light springs, and the disadvantage must chiefly appear in the inertia of this weight E.

In the paper under consideration the author mentions the possible disadvantage of the inertia of this weight, but thinks it more than overcome by the stability it gives against disturbances due to varying resistance of the valve gear; or, to use the author's words, "heavy weights are therefore an advantage instead of a detriment, as they prevent the valve from controlling the governor by their inertia and produce a governor strong enough to overcome any valve resistance." This stability, the author states, is due to the inertia of the weights, and therefore a governor having

the weight E would actuate a valve gear having greater resistance than could be overcome by the hypothetical governor used for comparison, although both are supposed to have corresponding and equal governing forces. The inertia of this weight prevents its orbit of rotation from being disturbed by intermittent and varying impulses of resistance from the valve gear, but it also prevents prompt changes of orbit when the conditions require them, and when radial motion takes place it over-reaches the mark. Aside from the necessities of overcoming valve resistance, it would seem that the less inertia in the moving parts of a governor the better its performance, just as with a steam-engine indicator. In both cases the inertia of moving parts is unfavorable to perfect action. It is manifestly impossible to prevent an appreciable amount of inertia in centrifugal governors, because centrifugal force implies a mass of some magnitude moving in a path around a centre. The addition of further weight used as centripetal force instead of springs would seem to be objectionable because of the inertia, unless the requirements of valve resistance prevent the use of the more desirable spring force.

Other methods of overcoming valve resistance are found in the various interlocking devices interposed between the flying weights and the eccentric. These are often open to the objection of increased friction, or the "friction of rest," as the author describes it. It is perhaps possible to annihilate this friction, however, by the valve resistance, and hence obtain a frictionless governor, with inertia reduced to a minimum. The conditions most favorable to this would be with a valve whose minimum resistance was a considerable amount, and with a simple interlocking device barely sufficient to prevent the maximum valve resistance from disturbing the orbit of weights. The valve resistance being periodically in opposite directions changes the contact of surfaces in interlocking device from one side to the other twice in each revolution, thus keeping the mechanism alive, and destroying the "friction of rest." At the same time a slight impulse is conveyed to the weights favorable to radial motion twice in both directions each revolution. The weights, taking advantage of these favorable impulses, respond quickly to any unbalanced condition of the opposing governing forces, and the device becomes, in effect, entirely free from friction, and at the same time free from unnecessary inertia in its moving parts.

In theory this arrangement seems to promise more perfect

results than the use of the weight E, but whether this is realized in practice or not depends on the possibility of fulfilling the conditions.

Prof. D. S. Jacobus.—The problem of the shaft governor has not been discussed by mathematicians thoroughly, because to do this in an exact way involves complications which make it a very difficult one. Mr. Smith has solved the problem in an approximate way and has made a number of governors, and found that by adjusting the same he is able to obtain the degree of regulation which is indicated by his theory. I consider this to be a very good feature. There might be a criticism raised that other governors which do not involve so many scientific points can also be regulated within 1%. In the valve gear of a Buckeye engine there is a great deal of friction caused by revolving the sleeve on which the cut-off eccentric is placed, and nevertheless, in the case of our shop engine, this governor has been made to regulate within 1%. If this problem is taken up exactly the action of the centrifugal force on the springs has to be included, together with its action on the other masses. This is a very complicated problem, provided the spring is set at an angle to the radius; the spring then forms a sort of centrifugal catenary, a mathematical discussion of which was given by Prof. Webb at the Toronto Meeting of the American Association for the Advancement of Science. I think we might hear from Prof. Webb on this point.

Mr. D. W. Robb.—I have been very much interested in the papers and discussion on shaft governors, having recently had some experience in that line. Although I did not have time to examine the papers as carefully as I would like, or to bring any data, I have hurriedly made some memoranda on the subject which possibly may throw a little more light on the question involved in Mr. Smith's governor, although perhaps nothing more than is conveyed in the paper.

I was very well pleased also with the points brought out in the papers by Prof. Sweet and Mr. Armstrong. I think Mr. Smith's governor contains a valuable feature not found in many governors, although I cannot see that a governor having a portion of the centripetal force supplied by the centrifugal force of weights, instead of getting the whole of it from springs, would be more stable than one having the whole centripetal force from springs alone. But I do think there is a decided advantage in being able to relieve the springs of part of their work and to bring their tension or compres-

sion as far as possible within the safe working limit. I believe it is not the first time that the principle in Mr. Smith's governor has been made use of in shaft governors. For instance, the latest form of governor of the straight-line engine, if I understand it right, has this principle applied in a different way. but like many other points involved in a shaft governor, one or two of which have been very happily made clear by the papers before this meeting, this principle is perhaps not generally understood. A short time ago I designed a governor of the ordinary type, but with a flat leaved spring, such as is used in the straight-line engine. The spring was curved to the shape of the wheel and bolted to it, the other end being attached directly to the weight, arranged so that the centrifugal force of the fly-wheel pushed the spring away from the centre of the wheel. In this case it will be seen that the centrifugal force of the spring itself, which was considerable, was acting in opposition to the force of the spring and in unison with the fly-weights, which rendered it necessary to have a spring with an initial tension corresponding to the centrifugal force of the fly-weights plus the centrifugal force of the spring, of course neglecting leverages and other constructive considerations, thus causing the spring to come very near its safe working limit. If I increased the strength of the spring by making it longer and stouter, I also increased its weight, and, consequently, its centrifugal force. If I decreased the fly-weights so as to require less spring, I required a proportionately weaker spring. After experimenting with this governor I designed another with the leaf-springs arranged to pull toward the centre, when the weights would fly out. Thus the centrifugal force of the springs was brought in unison with the tension of the springs, and I found it possible to have much less initial tension on the springs than in the other design. Neglecting the difference in centrifugal force due to the movement of the spring, which involves a principle in itself which I have not time to explain just now, but which I hope to do at some future time, the general principle is that the spring must be designed to have a deflection of so many inches to the ton to correspond with the variation of the centrifugal force of the fly-weights due to their movement; and the advantage of having a counter-centrifugal force in the springs or in separate weights is that you may have your springs working, say, for illustration, from 0 to 500 lbs., while in the other case you would have the same character of spring working, say, from 500 to 1,000 lbs. tension, which will obviously carry the spring nearer the

safe working limit. In using spiral springs the springs may be kept within the safe working limit by making them longer and stouter, but in shaft governors, as ordinarily designed, it is difficult to get room for a long spring.

Mr. Jesse M. Smith.—Prof. Sweet states that proposition three is not true, and then goes on to describe how he takes one half of the double governor away and puts something else in its place to maintain the balance, so that the force of the proposition still remains. The professor does not state what part of my mathematics he does not understand, so that I am at a loss where to look for the error. It is evident that if the weight E has sufficient centrifugal force to resist W unaided by the spring at any given number of revolutions, it will do so at any other number; but it so happens that even if E and W are in equilibrium they are in unstable equilibrium, and the weight of W, if it starts to go out, will go to its extreme distance from the centre of rotation. Now the valve resistance is not a constant quantity, but changes from zero to a maximum and back to zero twice every revolution, so that the weight W has an opportunity twice in each revolution to be started outward by the valve resistance. Suppose the weight W in its inner position and the tension of the spring equal to zero, and the weights W and E calculated to be in equilibrium with the *mean* valve resistance; now, as the eccentric passes its dead centres the valve resistance has no effect, and W, being overbalanced, starts outward; but the instant it starts its centrifugal force increases, because the weight is farther from the centre, and at the same time E is forced inward and its centrifugal force is decreased, because it is nearer the centre. The spring now begins to act to restore the disturbed equilibrium, and as soon as W has left its inner position (which corresponds to the latest cut-off) the weights, valve resistance, and all are under the control of the spring.

If the four weights were concentrated in one, as suggested by Prof. Sweet, their inertia might be equally or more effective in resisting the disturbing influences of the valve; but what of the spring to resist the centrifugal force of this consolidated weight? There would be difficulty in finding a fly-wheel large enough to contain it. A forcible reason for using a double instead of a single governor is that in the double governor gravity has no effect, because the weights are always in balance about the shaft. The effect of gravity in disturbing a governor increases as the speed decreases, and so long as Prof. Sweet applies his single governor

to high-speed engines he will ave less of the effect described in his paper, No. CCCCVII., and more as speed decreases, but if he uses the double governor and counterbalances the eccentric, gravity will have no effect at any speed.

Mr. E. J. Armstrong thinks that a small amount of friction is a good thing. May it not be possible that in his experiments with governors having anti-friction devices there still remained considerable friction? If there were no friction in any of the joints of the governor, a very small force would tend to move the weights, and would move them if it had time. When the governor is revolving at constant speed the only variable force is the valve resistance, and it acts always in the same direction relative to the governor, but with a variable quantity. If the number of revolutions is small this valve resistance will have time to act and will move the weights, but if the speed is increased the valve resistance will act to "jerk" the weights in or out; but the weights will resist this action by their inertia, and will resist the harder the quicker the "jerk" and the greater the weight. Hence I say that heavy weights are an advantage and an element of stability. On the other hand, if there be no friction in the joints of the governor the slightest change in speed will suffice to cause the governor weights to take a new position. In other words, the weights are ready to move under the slightest force if this force be applied slowly, but they will resist a large force if applied quickly. But a small force applied slowly produces slow motion, and slow motion means small force of inertia even with heavy weights. It therefore appears to me that a governor having the least friction in its joints is in better condition to respond quickly and accurately, and with a small change of speed, than one having more friction, particularly if the amount of friction is not under control, as it is not in most cases.

Mr. Armstrong sums up as follows: "Only an amount of fly-weight which would balance the spring can be taken as having anything to do with shifting the eccentric, and this amount varies with the *position of the fly-weights;* the remainder of the weight is simply inertia weight." The weights W and E are tied together so that one cannot move without the other, and they both have an effect. Suppose the fly-weight W in its inner position, and by some means during an increase of speed of 1% the weight arrives in its outer position. By reference to the figures given near the end of the paper it will be seen that the moment of W in its inner position is 1,850 inch pounds, while in its outer position

the moment is 2,930. The difference is 1,080. The moment of E corresponding to the inner position of W is 1,744, and that for the outer position of W is 1,831, the difference is 93. W in its inner position is nearly balanced by E, there being a difference between the moments of the two of only 19 inch pounds; whereas for the outer position of W the difference in moments is $2,930 - 1,744 = 1,186$, and this last figure just about represents the moment of the spring, which for the inner position of W had no moment.

Now suppose W in its outer position, and that E be removed entirely, and W be decreased until it is just balanced by the spring, the moment of which is 1,186. If now the weight W returns to its inner position, the spring has lost all its tension. What will balance the moment of W in this position? It is evident there is nothing to balance it unless the spring has an initial tension, and it is just this initial tension which is supplied by the weight E. In other words, the use of the two weights W and E permits the realization of an ideal governor in which there is a single weight travelling from the centre of the shaft to a point outside of the centre, and which is balanced by a spring having no initial tension. In actual practice the weight W does move out, and it moves out with an increase of speed. If W has a moment of 1,850 in its inner position, and 2,930 in its outer position, why is the difference of these two figures not the measure of the strength of the governor, without regard to whether the centripetal force is supplied partly by E or wholly by a spring? The fact seems to have been lost sight of, that whenever W gains centrifugal force, E loses, so that W becomes relatively stronger as it moves outward.

Prof. Thurston asks for particulars of performance.

Mr. F. E. Idell gives some figures on three engines driving electric street-cars. No service is more severe on an engine and its governor than that required by electric cars. The load is never twice alike, the changes are sudden, from one extreme to the other, and these changes may all take place in a single minute. To meet these conditions and give a service which shall be satisfactory, and continue to satisfy during a long run, a governor must meet the four propositions stated at the beginning of the paper, and also meet the fifth, which Prof. Thurston has kindly added to the list, and which is: That the governor may act not only promptly and powerfully, but with *accuracy*, it must include an adjustment or special mechanism which shall insure quick movement of its parts into the required position, instant by instant. A governor free from friction will

meet this last proposition, for the reason that there being no friction to resist a change of position, the parts will tend to move with the slightest change in speed and will move promptly, but slowly, so that there will be no excessive forces of inertia developed, and the parts will take their new positions accurately and will hold them firmly if there be sufficient mass in the governor-weights. I believe, therefore, that if a governor has weights of sufficient mass, and friction is reduced to minimum, it will be powerful to overcome valve resistance, and it will act promptly, accurately, and firmly without a dash-pot or other similar mechanism.

As regards my own practical results with this governor, I will state that the first engine I built was 12 x 14, which ran at about 200 revolutions per minute, with 90 lbs. steam pressure, and drove an old-style 40-light Brush arc-light dynamo, where every light meant a horse-power. There was a main valve driven by a fixed eccentric with a "gridiron" cut-off valve on its back, both unbalanced. The cut-off valve was driven directly from the eccentric, which was controlled by the governor, substantially as shown in Figs. 228, 229. It was with this engine, after trying several springs and with a large number of different tensions on the spring, that I found that the initial tension could be reduced to zero; and in order to get both springs adjusted alike, I screwed the washers back till they just touched the springs and so that I could compress them out of contact with the washers by my fingers. The engine drove this dynamo and nothing else for several months with entire satisfaction, and I saw the electric circuit broken and remade a number of times in quick succession, thus throwing on and off instantly 40 horse-power without a change in speed appreciable to the eye or ear.

Another engine, 10 x 12, of similar design, drove a factory. It ran at 150 revolutions per minute. With a constant load, the steam pressure being at about 90 lbs. the fire was banked and the pressure ran down to about 30 lbs. with a loss of speed of less than two revolutions.

Mr. Ball thinks it may be possible to annihilate the friction of the governor by the valve resistance. If the valve resistance be large enough to keep the parts in constant motion, or "alive," it would seem that it would enter as one of the important forces among the several which go to make up the equilibrium in the governor. This valve resistance, I think, acts, relatively to the parts of the governor, always in the same direction and not in

opposite direction, as stated by Mr. Ball. It is very irregular and uncontrollable, being dependent on the amount and quality of cylinder oil used, condition of valve seats, amount of water entrained by the steam, and many other things. It seems to me this is introducing one bad and uncertain element to take care of another just as bad.

Prof. Jacobus speaks of an engine in which the eccentric is mounted on a sleeve, and says the engine has been made to regulate within 1%, although there is a great deal of friction. This is very possible so long as the friction remains constant, but what happens when the sleeve begins to get dry during a long run?

In regard to the centrifugal force of the spring itself and its effect to distort the spring, as referred to by Prof. Jacobus and Prof. Webb, I have simplified this part of the question by placing the centre line of the spring on a radius so that its weight goes to form part of the weight E, and there is no effect of lateral distortion.

[NOTE.—This paper received discussion jointly with papers on the effect of an unbalanced eccentric, by Prof. J. E. Sweet (No. 407), and on a use for inertia in shaft governors, by Mr. E. J. Armstrong (No. 408). There are, therefore, references in the debate printed with each paper to some of the points brought out by the others.—*Secretary.*]

CCCCX.

TOPICAL DISCUSSIONS AND INTERCHANGE OF DATA.

XXIst. Meeting, Cincinnati, May, 1890.

No. 410–79.

Does a boiler steam more freely if the tubes are arranged so as to be farther apart, horizontally, in the upper rows than in the lower rows?

Mr. John M. Sweeney.—It is probable that I am to a certain extent responsible for the questions being in this place. Some few months ago it occurred to me to place the flues in a tubular boiler with an increasing cross-sectional area between them, and before working on the idea I suggested the propriety of bringing the question before the Society. In the meantime a boiler on this principle has been constructed, though I have not been able to make any exhaustive test of it. The flues as usually placed in the boiler now are, perhaps, one inch apart, and in some cases with an increased space between them at the centre of the boiler. The idea of the design presented is to increase the space between the flues as the flues rise. A point is taken outside the shell of the boiler, from which lines are drawn radiating from that point as a centre, and diverging, of course, as they proceed, and on these as centre lines the flues are placed (Fig. 245). The reason for this construction is that in a boiler externally fired the cross-sectional area which is found between the lower row of tubes has

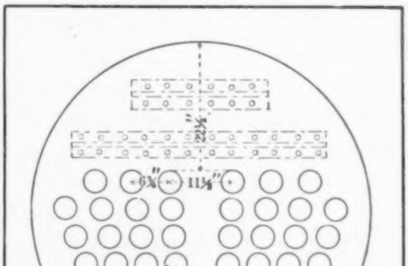


Fig. 245.

a duty to perform which consists in allowing the steam which is generated by the part of the shell below them and by the bottoms of the flues—to allow that amount of steam to ascend, and the water to replace it to descend. Now, if that inch of space between flues in this lower row is of a proper area for the duty which is to be performed by that row of tubes, certainly it is an improper area for the cross-section which is to be performed by the top row of flues, for they have not only the same duty to perform, but in addition to allow the passage of the steam which is generated in the part of the shell between the lower and the upper row and by the intermediate tubes. So that it seems to be reasonable that the cross-sectional area between the flues should be proportioned to the work to be done. The idea was not original with me at all, but came into my mind from some suggestions which were made in a book of C. Y. Williams on Heat, in which the axiom is laid down that it is more necessary to have the plates of the boiler swept by the water in circulation, in order to absorb the heat which is applied to the surface, than it is to apply the heat to the outside of the boiler. Without having made any tests whatever I would say that the working of that boiler so far has been notoriously free from any fluctuation in the water-line. It is very steady in its operation and seems to steam very freely.

Mr. J. Wendell Cole.—Would it not be better than to have all the tubes in the boiler of the same diameter and on radial lines, to have the tubes in the boiler on vertical lines parallel, but allow the lower lines of tubes to be of larger diameters and the upper lines of tubes of smaller diameters than the mean, thus giving equal centre spacing and yet not interfering with the rising of the steam generated between the vertical lines?

Mr. H. H. Suplee.—I understand that some boilers have been made according to the plan suggested by Mr. Cole, not only to give more space for circulation, but to equalize the draught through the tubes. The heated gases tend to rise and pass more freely through the upper rows, but by making the lower tubes larger, a greater volume of heated gases passes through this part of the boiler, while doubtless circulation is also helped in the way suggested by Mr. Sweeney. Two such boilers were built in Philadelphia, and, while I cannot submit data of test, I know they have given excellent and satisfactory results.

Mr. C. S. Dutton.—The arrangement of tubes as described by Mr. Sweeney has been used for quite a number of years by

Mr. T. R. Butman in boilers which he has built; and if I am not mistaken several years ago he obtained a patent for that arrangement.

Mr. A. F. Nagle.—The illustration represents the tubes radiating from a centre somewhat below the outside of the shell in a horizontal boiler, to facilitate the escape of steam from the tube surfaces.

It resembles a vertical boiler made by Mr. Edwin Reynolds, member of this Society, where the tubes are arranged in the same manner to facilitate cleaning of the tubes through a manhole in the shell.

Mr. D. L. Barnes.—In a boiler designed to procure the freest circulation of water, the point of the application of feed-water should be considered. Supposing that to be a stationary boiler fed in the centre; in that case the feed-water passes downward between the tubes, and the steam would probably rise around the shell at that end of the boiler where it is fed.

Prof. J. B. Webb.—I do not know exactly that I understand what is meant by steaming freely. There are three points to consider; in the first place, the amount of steam made; in the second place, the dryness of the steam; and in the third, the circulation, to prevent there being any place in the boiler where dirt might settle, due to any eddy or quiet place in the boiler. As to the amount of steam made, it seems to me that as long as the flues are kept wet the same amount of steam must be made, so that the whole matter of the arrangement of the tubes would seem to be governed by other considerations. A poor circulation may induce priming, by making the water-level too high, and it should be remembered that the water-level shown in the tube is not that in the boiler. By steaming freely I do not know what is meant, unless it be to make more steam.

Mr. Sweeney.—I have taken it for granted that the top feed in the boiler is the accepted feed, but it strikes me that no matter where it be fed the water is circulated up through the tubes and has to come down again. The advantage, as I understand, of free circulation in a boiler is that water agitated over the heating surface will take up heat more rapidly than if it is not kept moving. I know that in a coil boiler, that known as the Ward boiler, very similar to the Herreshoff, it is claimed that water passes in circulation through a tube, and being carried past an opening above, will give its steam off, and that very little steam room is

required in order to secure that all the steam is eliminated from the water.

No. 410-80.

What advantages as to construction and operation are offered by vertical bending rolls over horizontal rolls for plate?

Mr. Jas. W. See.—While probably not exactly pertinent to the question, I want to call attention to a point of advantage which one user has found in horizontal rolls, and would like to ask if it is thought to be legitimate. A party has a set of bending rolls for bending tank iron or tank steel half an inch thick; and the rolls were built for that purpose and were intended to bend to 12 or 14 inch radius. The parties had in their shop an overhead crane, a very convenient thing for handling those plates, and the way they bend the corners of their tanks is to run those plates in their rolls and come down with the top roll and hitch on the travelling crane and just snake that plate straight up. It breaks the rolls all to pieces, but the maker of the rolls has to stand it. I want to know if that roll thus used is a bending roll or a mere brake?

Mr. Oberlin Smith.—I was quite interested while in Liverpool last year in looking at some vertical bending-rolls at the shops of the Messrs. Laird, where they were bending plates one and a half inches thick and perhaps ten feet square. The rolls seemed to be capable of even heavier work. The plate rested upon its edge, and one advantage of this system would seem to be the saving of floor room. I suppose it is just as easy to hoist the plates up by one edge as to handle them flatwise. Another advantage is that the plate, by gravity, tends to remain true in the rolls. With the horizontal system, on the contrary, there is some tendency to creep laterally one way or the other, which has to be controlled by flanges on the rolls, or other guides.

Mr. Jno. B. Crocker.—I would like to ask if there is any difficulty in making conical sheets with that system?

Mr. Oberlin Smith.—With the vertical system that I speak of I do not think conical sheets could so well be bent. It seemed to be best calculated for cylindrical work.

No. 410-81.

What is the best form of hydraulic valves for water at high pressures?

Mr. G. C. Henning.—I should like to say that in experience with high pressure of water I have found one valve which acts

equally well on a swivel joint or a fixed joint. It never leaks under different temperatures, under varying pressures up to 4,500 pounds to the square inch, and, of course, acts equally well on low pressures. All other valves which I have thus far seen invariably leak. This valve is the A. H. Emery valve, such as is used on the Emery testing machines (Fig. 246). It is equally applicable to other work. It is made of a special bronze called the Emery bronze, which has a tenacity of about 80,000 pounds to the square inch. The valve seats are prepared by hammering the metal, then turning them down generally by dies; the valve is formed by a little plug, generally a high-steel plug, which is carefully turned and ground to fit the valve seat. The shape of this valve seat is conical for a very small part of the opening, and then it curves back so as to give plenty of clearance, that any dirt, etc., can get into the valve without sticking under the seats readily. I have used these valves for some time and know they are very good. Some have been used up to 10,000 pounds per square inch without leaking.

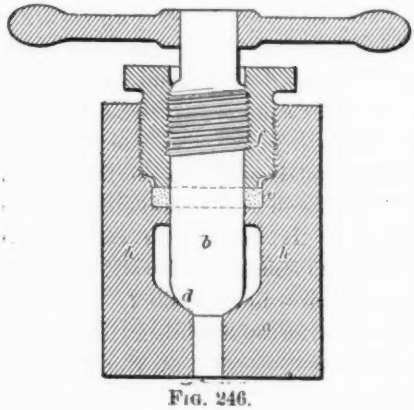


FIG. 246.

Mr. J. H. Samuels.—I would like to ask the gentleman who just spoke how fast these valves can be worked.

Mr. Henning.—These valves can be worked as fast as you can put mechanism to the upper end to raise or lower, though, of course, a valve under high pressure should never be opened suddenly. You always must have some device for holding the valve down and some means of opening it—generally a hand-wheel and screw, never a lever. As fast as you can operate such mechanism the valve can be opened.

Mr. Louis S. Wright.—This discussion about hydraulic valves has brought to my mind a valve on a cast-iron main from an accumulator, the valve being located where the water is drawn off from the main, for tools, etc. The working of this valve having been satisfactory, I thought it might be of interest here, and, if I am allowed to make a sketch, I will detail its particular features.

The shell of the valve is of cast iron, and is cast fast to a short section of the main from the accumulator. The valve stem is brass. The essential features of this valve are: First, it can be packed without shutting down the water pressure in the main and thus throwing out of service all other machines getting pressure water from the same main. Second, the construction is cheap, there being but a small amount of machine work on it. Third, ther is no seat which needs grinding to keep it in working order, and no flat surfaces to the seat between which small particles in the water can lodge and prevent the valve from closing.

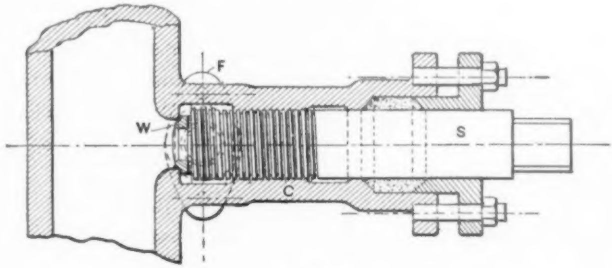


FIG. 249.

Referring to the sketch (Fig. 249), the cast-iron shell is shown here as marked as *C*. The stem *S*, of brass, has a square end to take a lever or hand-wheel. The inside end of the stem carries a small brass washer *W*, which is fitted loosely to the main valve stem *S*. Back of the valve, as here shown, is sketched a flange *F*, which is on a wrought-iron pipe leading from the valve to the machine to which the pressure water is to be led. This flange is bolted fast to the valve shell, making the joint between the valve and the wrought-iron pipe to the machine. The sharp corner left on the cast-iron shell, where the washer *W* is shown as touching that part of the shell, cuts into the brass washer when it is forced to its seat when the valve stem is screwed up, thus making a tight joint. Any matter lodging at the seat is forced to one side or the other of the sharp corner. This valve as sketched was used at 750 lbs. pressure, and was quite satisfactory. There is no method here to prevent the stem from blowing out if carelessly unscrewed too far, which in one instance did happen, and narrowly escaped injuring a man. To prevent this an encircling stop can be fastened to the shell, which will limit the throw of the valve stem.

I should like to ask if the Emery valves have any valve stems

which have to be packed, and if so, what means he takes to pack these valve stems, as that frequently causes trouble.

Mr. Henning.—Yes, sir; they require packing, of course. The packing is simply a leather ring. The valve seat is shown at *d* (Fig. 246). The stem rises, and the liquid, of course, comes out through *h*; there is a space all around the stem. This valve stem *b* must be packed. To pack it there is a recess with a leather packing ring *e*. The packing ring is held in place by the gland *f*. It has a screw thread to hold it in place. These packing rings have the thickness of one piece of leather, sometimes two. If there is the slightest leakage under *e*, a little pressure applied by screwing down the gland will make it perfectly tight. These valves are generally too small to use cupped-leather packing. In the cylinder of the testing machine there are cup-leather collars, and they are tight for a certain length of time. They do keep tight, when frequently using the testing machine, for six months or more. But after that there is no telling how long they will keep tight.

Mr. Nason.—What is the diameter of the inlet?

Mr. Henning.—The largest that I know of are an inch and a half valve. They are just about an inch opening.

Mr. Nason.—What was the thickness of the metal?

Mr. Henning.—About three-quarters of an inch in the thinnest part.

Mr. A. H. Raynal.—In trying some of our hydraulic machinery, recently purchased abroad, I frequently found the valves leaky. In correcting some of them I had to make both new casings and new seats, and in ignorance of Mr. Emery's successful special metal, I used Prof. Thurston's alloy. I was much pleased with the success of it. It was very dense and strong, and I have had no trouble with the valves since.

Mr. H. D. Hibbard.—I might say that the steel works of this country use hydraulic machinery very largely, and the valve which is in general use in them is the Critchlow valve, which answers most purposes very well indeed, and as the patent on it quite recently ran out, it seems to be a very good valve to use for hydraulic purposes.

Mr. C. M. W. Smith.—I would like to ask if the seat in the Emery valve is independent of the body, or is it a piece cast with the body?

Mr. Henning.—The lower end of the valve comes down onto the bearing. It is not screwed into the stem of the valve at all, but the

lower end of the stem is simply finished off. The main point is that the seat and valve be so correctly made that the shank will be accurately in line with the bearing and that there is no possibility of variation. Just as soon as you attach an end to that shank there it will be extremely difficult to get it exactly concentric with the bearing and with the seat. So long as that is not absolutely true you cannot have a tight valve.

Mr. C. M. W. Smith.—The question is in regard to the seat.

Mr. Henning.—The seat is a solid seat. It is one solid piece of metal. If it does begin to leak all that is necessary is to ream it out a little.

I have just been asked another question about the valves, and that is, why they should be tighter when the pressure runs higher. It is simply that the pressure under the valve in the tube forces the valve seat against the valve, and the higher the pressure is when you once have closed the valve, the tighter the valve will close.

No. 410-82.

Is it better or not to have the lead increase with the load in high-speed automatic engines, and if so, why?

Mr. W. O. Webber.—As I offered that topical question for discussion, I would say that I asked the question very largely for information, and I should like very much to hear what the other members who have had more experience than I have to say about it. What led me to propose this topic was this, that in testing very many high-speed engines for close regulation, with the shaft governor set so that the path of the eccentric or eccentric pin was squarely across the shaft, so that the lead did not increase, I found a greater variation in the revolutions between extreme high load and the engine running light than I did with an increased lead. I found it out by accident in testing an engine without the key inserted in the governor. This was a governor separate from the pulley. In making some adjustments the governor plate got moved ahead about one-eighth of an inch, and it showed immediately a much closer regulation, and we then attempted to find out whether that was due to the increased angular movement ahead or otherwise, and not fully convincing myself, I therefore asked the question.

No. 410–83.

Have you had any experience with systems for purification of bad feed-water before it gets into a steam-boiler, either by chemical precipitation or otherwise?

Mr. Ezra Fawcett.—We have been using a feed-water for the boiler at our works from an *artesian* well, which deposited a white substance in the boiler, but little hard scale. In order to purify and heat the feed-water as much as possible we constructed an open heater, which could be easily cleaned and attended, about two feet diameter by six feet high, with a perforated plate two feet from the bottom, with a suitable door above the plate to get inside and fill the space above the plate with foundry coke. We let the exhaust steam enter below the plate, pass through the coke, and the feed-water come in at top and pass down through the coke and exhaust steam. By this plan we precipitate much of the impurities of the feed-water, heating the feed to near the boiling point and with results which were very favorable to the coal pile.

Mr. John A. Laird.—We have had some experience in the St. Louis Water Works with the live-steam feed-water heater and purifier manufactured by Mr. Hoppes, of Springfield, O.

It has been in use for two years now, and has given very good satisfaction, though considerable more water is evaporated in the battery of four boilers to which it is attached than was originally intended. The water used is Missouri River water, which is very muddy and forms a hard scale which must be removed with the scaling tool.

The boilers which have the purifier on show a very small amount of scale near the point where the feed-water enters, and there is no doubt but that the purifier would catch all the scale in the water if it was not working above its rated capacity.

Mr. H. H. Suplee.—In some experiments which I made a few years ago, I found that the purifying capacity of a heater depends very much on its cubical contents, or, in other words, upon the length of time which it takes a given volume of water to pass through it. While the heating capacity of several heaters may be the same, yet in that through which the water passes rapidly much less of impurities will be deposited than in another, where the water can remain nearly stagnant or as in a tank. In this latter the carbonates and most of the clay and earths would settle and remain. The defect of most heaters results from the attempt to crowd too much heating surface into too small a volume, or within too restricted a shell.

No. 410–84.

What experience have you had with power-moulding machines in the foundry? What difference do they make in the output?

Mr. Edgar Penney.—I will give an experience of my own with a simple power-moulding machine. Previous to its use a man and boy put up but 33 snap-flasks a day.

The machine was made something after the style of an upright drill (Fig. 248), with a counterbalanced ram having a platen about

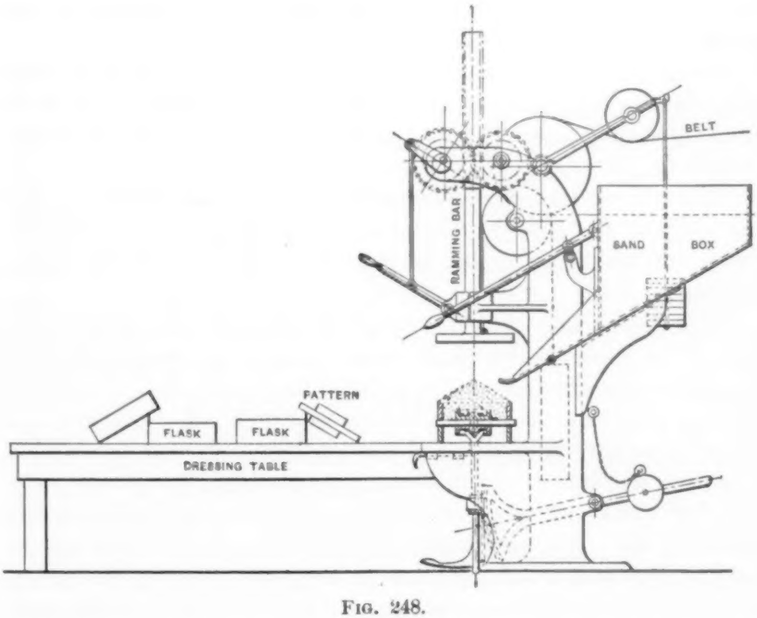


FIG. 248.

18 inches square on the lower portion of the same. This ram-bar was actuated by a pair of grooved friction-wheels applied very similarly to the lifting or upper works of a Merrill drop-hammer, by forcing a pair of revolving rolls against the grooved upper part of the ram-bar.

The flasks were provided with trunnions, the patterns upon plates.

A man and two boys have put up with this machine 166 12-inch snap flasks per day.

This increase in output was gained by the time saved by mechanical ramming and the facility with which the flask was turned and

manipulated, and by having a number of snap flasks and duplicate pattern plates.

The work as it came from the machine was passed along the bench to the moulder, who did the drawing of the patterns and the dressing, a boy carrying to the floor.

The flask used was about a 12-inch cast-iron open or snap style, and the work done, shield plates and small gears, in fact anything which could be put on the 12-inch plates, was successfully handled. The sand was simply shovelled or piled up on to the flask to any convenient height, no attention being paid to this feature so long as sand enough was provided.

The platen was then forced down on this conical heap of sand piled upon the pattern plate and flask, the result being much more even ramming than could possibly be done by hand bench moulding.

With two hand rammers moulders are apt sometimes to ram unevenly, and this is clearly shown in flat work; by exposing the casting to light at an angle, you can detect each blow of the hand-rammer.

With the machine the pressure of the platen on the cone of sand seemed to permeate the whole mass of sand and produced even pressures upon the face of the pattern.

After pressing, the surplus sand was struck off. I might add that the pressure upon the sand was regulated as near as may be, according to the area of the flask operated upon, by two methods. The first was to use the momentum of the driving fly-wheel and exhaust that. The next method and most successful one was to put an adjustable weighted idler on the belt, allowing the belt to slip at the moment the proper pressure to suit the area of the work was exerted upon the flask.

Mr. Nason.—Do we understand that the patterns were in halves and the two halves put on the opposite sides of the plate?

Mr. Penney.—The patterns were made in halves and mounted upon a plate. We will take, for instance, a pattern of a small truck-wheel, where the web of the wheel is in the centre.

This pattern was parted right through the middle, half of it put on each side of the plate.

The flasks at the joints were accurately planed, the plate fitting between the same, guided by dowels.

The manner of operating was to put the drag flask upon the fixed platen of the machine. The plate was then laid upon the

flask, fitting the dowel pins. Upon that was put the "cope" or upper half of the flask. A boy then filled the "cope" with sand, heaping it up. In fine casting the workman shook a handful of prepared dressing from a sieve over the pattern, sometimes tamping it to the surfaces with his fingers, then threw in the rough sand, applying the pressure, struck off the surplus sand, threw up a pair of hooks which locked the two flasks together and released a catch which allowed a pair of counter-weighted forked rods or bearers to move upwards, which caught under and carried the trunnion provided upon the plate or flask.

The flask was raised clear of the table; the "cope" being filled with sand, by its own gravity turned over. The workman pressed the lifting-rods back by applying his foot to a treadle. The filling and ramming being repeated and flask shoved along the bench to the moulder, the machine being kept continually at work, enough flasks and patterns being provided for this purpose.

The principal feature in this machine, aside from the application of power, was its convenient arrangement and its being adapted to suit all sizes of flask within its scope without special adjustment, and that no particular care need be exercised in the amount of sand which was piled upon the flask.

This machine would take anything from a flask an inch deep up to 24 inches high to 18 inches square.

Mr. Carleton W. Nason.—Do you use a pattern dropped through a plate, as is commonly done now?

Mr. Penney.—Some moulding was done in that way.

Mr. Nason.—I don't think you quite understand the method I mean. With the machine I have in mind one-half of a pattern was used. This was laid upon an iron plate, and after being marked out a hole was cut through the plate, through which the pattern could be dropped. The fit was necessarily a nice one, so that no sand could follow.

The pattern is then fastened to a stiff frame beneath the plate, which is capable of having a drop motion of from $\frac{1}{2}$ to $\frac{3}{4}$ inch which is given by any mechanical device, say either lever or cam.

Snap-flasks were used, as in the case which Mr. Penney cites. In starting to mould, a flask was placed on the plate, filled with sand, the latter being rammed up with a foot-motion and lever to any desired degree of firmness. The pattern was then dropped down through the plate, as stated, leaving the sand behind it, and the flask removed with the sand from the plate.

The cope was similarly made, and the second half being thus placed on the first, the mould was complete.

It is obvious that this form of machine was only adapted to classes of work where both sides of the pattern were alike; and there were adjusting-screws in each end of the plate in the machine for the purpose of adjusting the position of the flasks and the patterns, by means of the pins on it coming at the fixed places indicated by the screws.

After the mould was finished the flasks were both removed together and the process recommenced.

Gates were on the plate by the pattern, so that no time was lost in cutting them out of the sand afterward, and the mould was complete without "doctoring" after being removed from the machine.

The capacity of these machines with flasks about 10 × 14 was from 150 to 175 per day for each man, including the labor of pouring.

Mr. Penney.—Do you mean the use of what is called the "stripper"?

Mr. Nason.—I do not know it by that name.

Mr. Penney.—The method which you describe is what I would call using a stripper. This machine was sometimes used in that way. It is not every kind of pattern which can be moulded by using a stripper-plate.

I simply speak of this experience of mine, and am not interested in any moulding machines, to show the difference in the amount of work accomplished by power-moulding compared with hand, and account for it by attributing the saving to the time gained in ramming up and handling the sand, which was done by one operation—the fall of the power-platen did the ramming; also to the handling of the flask, which was much facilitated, and to the use of plate-moulding and division of labor, as it were.

I have knowledge of power-moulding machines in use in which effort is made to distribute the ramming pressure over the pattern, if it be of an irregular shape, by use of multiple rammers or yielding surfaces, the object of which is to keep an equal pressure on the sand to suit the varying depth of the sand or the irregularity of the patterns.

This refinement did not prove to be necessary on ordinary work in the flask of the size used.

Pile upon the flask sufficient sand, cone-shaped, let the pressure

come down firmly and gradually, and you will find with the ordinary pattern (there may be some very irregular patterns which could not be reached by this flat platen process) that the ramming is very even indeed.

There appeared to us no necessity for irregular, yielding, or multiple rammers to be used for flasks up to 18 inches square.

CCCCXI.

APPENDIX II.

*GENERAL SOLUTION OF THE TRANSMISSION OF FORCE IN A STEAM-ENGINE AS INFLUENCED BY THE ACTION OF FRICTION, ACCELERATION, AND GRAVITY.**

D. S. JACOBUS,† HOBOKEN, N. J.

THIS problem has been discussed by many, but the author has met with no perfectly general solution. Various approximate ones are employed in general engineering work involving errors of unknown magnitude. The author has, therefore, endeavored in this paper to present a general set of equations by means of which the errors of the more approximate ones may be ascertained.

The following general conditions have been assumed:

(*a*) That the centre of the crank shaft is not necessarily on the line of travel of the wrist-pin; ‡

(*b*) that the centre of gravity of the connecting-rod is not necessarily in its line of centres;

(*c*) that the crank revolves at a uniform speed;

(*d*) that the mass of the moving parts is distributed in any manner;

(*e*) that the line of travel of the wrist-pin is not necessarily horizontal or vertical;

(*f*) that there is friction between the parts.

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* This Appendix is added in response to the request made at the New York meeting, where the original paper was presented. See page 492 of this (XIth) volume. It is reprinted from the *Annals of Mathematics*, February, 1890, and was forced out of its place in the First Part by reason of the delay referred to on page 526.

† I am greatly indebted to Prof. J. Burkitt Webb for valuable assistance in preparing this article, many of the equations having been worked out independently by him, and compared in order to check the final result. The methods employed in the solutions have also been in a great measure suggested by him.

‡ Engines are now in use that require this condition to enter the equations; for example, the Westinghouse.

Fig. 251 represents the main lines of an engine referred to the horizontal axis OX, coinciding with the line of travel of the wrist-pin, O being the extreme point of its travel and OY the vertical axis; the directions of the various quantities are indicated by arrow-heads.

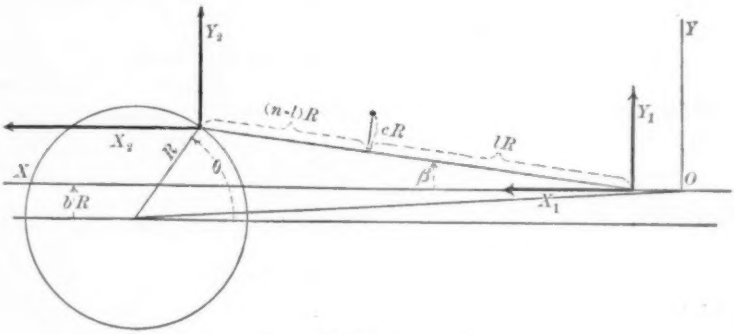


FIG. 251.

NOTATION.

Let F_1 = the force to accelerate the piston-rod, and cross-head;

X_1, X_2 and Y_1, Y_2 = the x and y components of the forces which accelerate the connecting-rod, applied at the wrist and crank-pins respectively;

R = radius of the crank;

nR = length of the line joining the wrist and crank-pin centres;

lR = distance from the wrist-pin to the foot of the perpendicular let fall from the centre of gravity of the connecting-rod upon this line;

cR = length of this perpendicular;

bR = perpendicular from the centre of the crank-shaft upon the line of motion of the wrist-pin;

θ = crank angle;

β = connecting-rod angle;

M = mass of piston, piston-rod, and cross-head;

m = mass of the connecting-rod, and

kR = its principal radius of gyration;

W_1 = weight of the piston, piston-rod, and cross-head;

W_2 = weight of the connecting-rod;

v = angular velocity of crank;

s = length of stroke;
t = time required by the crank to turn through the angle θ.

FORCE REQUIRED TO ACCELERATE THE PISTON, PISTON-ROD, AND CROSS-HEAD.

The general value of x, the co-ordinate of the wrist-pin, is

$$x = \sqrt{(R + nR)^2 - b^2 R^2} - R (\cos \theta + n \cos \beta). \qquad (1)$$

From the figure we have

$$\sin \beta = \frac{\sin \theta - b}{n}; \qquad (2)$$

therefore,

$$\cos \beta = \frac{1}{n} \sqrt{n^2 - (\sin \theta - b)^2}, \qquad (3)$$

which reduces (1) to

$$x = \sqrt{(R + nR)^2 - b^2 R^2} - R [\cos \theta + \sqrt{n^2 - (\sin \theta - b)^2}]. \qquad (4)$$

Differentiating, we have

$$\frac{dx}{d\theta} = R \left(\sin \theta + \frac{(\sin \theta - b) \cos \theta}{\sqrt{n^2 - (\sin \theta - b)^2}} \right),$$

and

$$\frac{d^2x}{d\theta^2} = R \left(\cos \theta - \frac{\sin \theta (\sin \theta - b)}{\sqrt{n^2 - (\sin \theta - b)^2}} + \frac{n^2 \cos^2 \theta}{[n^2 - (\sin \theta - b)^2]^{\frac{3}{2}}} \right).$$

For brevity, let

$$\frac{n^2 \cos^2 \theta}{[n^2 - (\sin \theta - b)^2]^{\frac{3}{2}}} - \frac{\sin \theta (\sin \theta - b)}{\sqrt{n^2 - (\sin \theta - b)^2}} = Z; \qquad (5)$$

then

$$\frac{d_2 x}{d\theta_2} = R (\cos \theta + Z). \qquad (6)$$

But

$$\tau = \frac{d\theta}{dt}, \qquad (7)$$

by means of which $d\theta$ may be eliminated from (6), giving

$$\frac{d^2x}{dt^2} = \tau^2 R (\cos \theta + Z);$$

from which it follows that

$$F_1 = M\tau^2 R (\cos \theta + Z). \qquad (8)$$

FORCES REQUIRED TO ACCELERATE THE CONNECTING-ROD.

The general equations for the forces to accelerate the connecting-rod have been derived by two methods:

1. By determining the forces at the centre of gravity required to vary the translation of the mass, and the moment necessary for its angular acceleration.
2. By assuming the mass of the rod to be symmetrically arranged, say in two equal and opposite portions, at a distance from the centre of gravity equal to the radius of gyration, and determining the forces and moments required for the acceleration of such a representative rod.

FIRST METHOD.

The motion of the connecting-rod is composed of a translation of its centre of gravity and a rotation of the rod about it, this translation and rotation being so related as to cause the wrist-pin to follow a right line, and the crank-pin a circle. This motion might be considered as a rectilinear translation of the wrist-pin and a rotation about it, but is not so taken, because the centrifugal forces of the parts of the rod are not balanced about that point, and complicate the problem.

To determine the forces at the wrist- and crank-pins required to accelerate the centre of gravity, we determine the force necessary when applied at that point, and resolve it into two equivalent forces applied at the wrist- and crank-pins, which latter forces will be inversely proportional to their lever arms about the centre of gravity. We thus obtain forces at the pins that will produce translation only.

To determine the rotative forces we find the moment necessary to produce the rotation, and then place such equal and opposite forces at the pins as will produce it.

The sum of the translative and rotative forces at the pins will be the total forces required to produce the complete acceleration of the rod.

Let x and y be the co-ordinates of the centre of gravity (see Fig. 251); then we have

$$x = \sqrt{(R + nR)^2 - b^2 R^2} - R\,[\cos\theta + (n - l)\cos\beta + c\sin\beta].$$

Substituting the values of sin β and cos β from (2) and (3), we obtain

$$x = \sqrt{(R + nR)^2 - b^2R^2} - R\left(\cos\theta + \frac{n-l}{n}\sqrt{n^2 - (\sin\beta - b)^2} + \frac{c(\sin\theta - \beta)}{n}\right)$$

from which

$$\frac{dx}{d\theta} = R\left(\sin\theta + \frac{(n-l)(\sin\theta - b)\cos\theta}{n\sqrt{n^2 - (\sin\theta - b)^2}} - \frac{c\cos\theta}{n}\right),$$

and

$$\frac{d^2x}{d\theta^2} = R\left[\cos\theta - \frac{(n-l)(\sin\theta - b)\sin\theta}{n\sqrt{n^2 - (\sin\theta - b)^2}} + \frac{(n-l)\,n\cos\theta}{[n^2 - (\sin\theta - b)^2]^{\frac{3}{2}}} + \frac{c\sin\theta}{n}\right].$$

Introducing the values of Z and t from (5) and (7), we have

$$\frac{d^2x}{dt^2} = \tau^2R\left[\cos\theta + \frac{n-l}{n}Z + \frac{c}{n}\sin\theta\right].$$

Let F_x be the x component of the translative force at the centre of gravity; then

$$F_x = m\tau^2R\left[\cos\theta + \frac{n-l}{n}Z + \frac{c}{n}\sin\theta\right]. \qquad (9)$$

From Fig. 251 we have for the ordinate of the centre of gravity

$$y = lR\sin\beta + cR\cos\beta = \frac{lR}{n}(\sin\theta - b) + \frac{cR}{n}\sqrt{n^2 - (\sin\theta - b)^2}.$$

Differentiating, we have

$$\frac{dy}{d\theta} = R\left(\frac{l}{n}\cos\theta - \frac{c(\sin\theta - b)\cos\theta}{n\sqrt{n^2 - (\sin\theta - b)^2}}\right),$$

and

$$\frac{d^2y}{d\theta^2} = -R\left(\frac{l}{n}\sin\theta - \frac{c(\sin\theta - b)\sin\theta}{n\sqrt{n^2 - (\sin\theta - b)^2}} + \frac{cn\cos^2\theta}{[n^2 - (\sin\theta - b)^2]^{\frac{3}{2}}}\right);$$

from which

$$\frac{d^2y}{dt^2} = -\tau^2R\left(\frac{l}{n}\sin\theta + \frac{c}{n}Z\right).$$

Let F_y be the Y component of the translative force at the centre of gravity; then

$$F_y = -m\tau^2R\left(\frac{l}{n}\sin\theta + \frac{c}{n}Z\right). \qquad (10)$$

In order to determine the rotative forces, we proceed as follows: From (2) we have

$$\frac{d\beta}{d\theta} = \frac{\cos\theta}{n\cos\beta},$$

$$\frac{d^2\beta}{d\theta^2} = -\frac{\sin\theta}{\sqrt{n^2-(\sin\theta-b)^2}} + \frac{\cos^2\theta\,(\sin\theta-b)}{[n^2-(\sin\theta-b)^2]^{\frac{3}{2}}};$$

from which it follows that the moment required to produce the rotation will be

$$M_r = -mr^2R^2k^2\left(\frac{\sin\theta}{\sqrt{n^2-(\sin\theta-b)^2}} - \frac{\cos^2\theta\,(\sin\theta-b)}{[n^2-(\sin\theta-b)^2]^{\frac{3}{2}}}\right). \quad (11)$$

Let D_w, D_c and E_w, E_c be the horizontal and vertical forces acting at the wrist- and crank-pins, respectively, into which the forces F_x and F_y are divided; then we have

$$D_w nR\sin\beta = F_x\,[(n-l)\,R\sin\beta - cR\cos\beta],$$

and

$$D_c nR\sin\beta = F_x(lR\sin\beta + cR\cos\beta);$$

from which

$$D_w = F_x\left(\frac{n-l}{n} - \frac{c}{n}\cot\beta\right) = F_x\left(\frac{n-l}{n} - \frac{c\sqrt{n^2-(\sin\theta-b)^2}}{n\sin\theta-b}\right), \quad (1$$

and

$$D_c = F_x\left(\frac{l}{n} + \frac{c}{n}\cot\beta\right) = F_x\left(\frac{l}{n} + \frac{c\sqrt{n^2-(\sin\theta-b)^2}}{n(\sin\theta-b)}\right). \quad (13)$$

Dividing the vertical force F_y between the wrist- and crank pins in the same way, we have

$$F_y\,[(n-l)R\cos\beta + cR\sin\beta] = E_w nR\cos\beta,$$

and

$$F_y(lR\cos\beta - cR\sin\beta) = E_c nR\cos\beta;$$

from which

$$E_w = F_y\left(\frac{n-l}{n} + \frac{c\,(\sin\theta-b}{n\sqrt{n^2-(\sin\theta-b)}}\right), \quad (14)$$

and

$$E_c = F\left(\frac{l}{n} \quad \frac{c(\sin\theta-b}{n\sqrt{n^2-(\sin\theta-b)^2}}\right). \quad (15)$$

The value of M_r given in (11) may be written

$$M_r = m\tau^2 R^2 \frac{k^2}{n^2} Zn \sin\beta - m\tau^2 R^2 \frac{k^2}{n^2} \sin\theta \,.\, n\cos\beta\,;$$

therefore the rotation may be produced by two equal and opposite forces in the X direction at the wrist- and crank-pins equal to $\pm\, m\tau^2 R \frac{k^2}{n^2} Z$, together with a second set applied in the y direction at the same points equal to $\pm\, m^2\tau R\, \frac{k^2}{n^2} \sin\theta$.*

Adding these forces to those required to produce the translation, we have the total accelerating forces,

$$Y_1 = E_w + m\tau^2 R \frac{k^2}{n^2} \sin\theta \tag{16}$$

$$Y_2 = E_c - m\tau^2 R \frac{k^2}{n^2} \sin\theta, \tag{17}$$

$$X_1 = D_w + m\tau^2 R \frac{k^2}{n^2} Z, \tag{18}$$

$$X_2 = D_c - m\tau^2 R \frac{k^2}{n^2} Z. \tag{19}$$

Substituting the values of E_w, E_c, D_w and D_c as given in equations (12) – (15), in equations (16) – (19) the accelerating forces become

$$Y_1 = -\, m\tau^2 R \left[\left(\frac{n-l}{n} + \frac{c\sin\theta - b)}{n\sqrt{n^2 - (\sin\theta - b)^2}}\right)\left(\frac{l}{n}\sin\theta \pm \frac{c}{n} Z\right) - \frac{k^2}{n^2}\sin\theta\right], \tag{20}$$

$$Y_2 = -\, m\tau^2 R \left[\left(\frac{l}{n} - \frac{c\,(\sin\theta - b)}{n\sqrt{n^2 - (\sin\theta - b)^2}}\right)\left(\frac{l}{n}\sin\theta + \frac{c}{n} Z\right) + \frac{k^2}{n^2}\sin\theta\right], \tag{21}$$

$$X_1 = m\tau^2 R \left[\left(\frac{n-l}{n} - \frac{c\sqrt{n^2 - (\sin\theta - b)^2}}{n\,(\sin\theta - b)}\right)\left(\cos\theta + \frac{n-l}{n} Z + \frac{c}{n}\sin\theta\right) + \frac{k^2}{n^2} Z\right], \tag{22}$$

* This division of M_r reduces the values of the accelerating forces to a form in which the value of Z may be employed, and numerical computations thus facilitated.

$$X_2 = mr^2R\left[\left(\frac{l}{n} + \frac{c\sqrt{n^2 - (\sin\theta - b)^2}}{n(\sin\theta - b)^2}\right)\right.$$
$$\left.\left(\cos\theta + \frac{n-l}{n}Z + \frac{c}{n}\sin\theta\right) - \frac{k^2}{n^2}Z. \quad (23)\right.$$

SECOND METHOD.

Fig. 252 represents the main lines of an engine in accordance with the second supposition; kR, kR are two radii of gyration drawn from the centre of gravity parallel to the line AB joining the centres of the crank- and wrist-pins, and the mass is supposed to be concentrated in two equal portions at their extremities D and E. This reduces the consideration of the motion of the rod to the more simple discussion of that of the two concentrated masses.

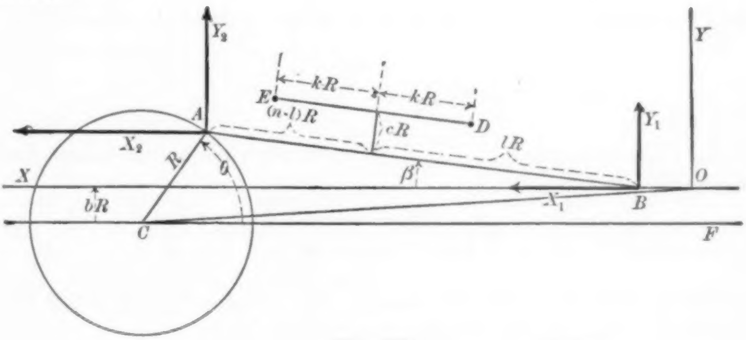


Fig. 252.

Let the co-ordinates of the masses D and E be x_1, y_1 and x_2, y_2, respectively; then we have

$$x_1 = C_1 - R[\cos\theta + (n - l + k)\cos\beta + c\sin\beta],$$

in which C_1 is the distance from the centre O to the ordinate of D when the crank is on the inner dead centre.

Differentiating the above value of x_1, we have

$$\frac{d^2x_1}{d\theta^2} = R\left[\cos\theta + (n - l + k)\left(\cos\beta\frac{d\beta^2}{d\theta^2} + \sin\quad\frac{d^2\beta}{d\theta^2}\right)\right.$$
$$\left. + c\left(\sin\beta\frac{d\beta^2}{d\theta^2} - \cos\beta\frac{d^2\beta}{d\theta^2}\right)\right]. \quad (24)$$

It also follows from Fig. 252 that

$$\sin\theta - b = n\sin\beta,$$

from which

$$\frac{d\beta}{d\theta} = \frac{\cos\theta}{\sqrt{n^2 - (\sin\theta - b)^2}}, \tag{25}$$

and

$$\frac{d^2\beta}{d\theta^2} = \frac{\cos^2\theta(\sin\theta - b)}{[n^2 - (\sin\theta - b)^2]^{\frac{3}{2}}} - \frac{\sin\theta}{\sqrt{n^2 - (\sin\theta - b)^2}}. \tag{26}$$

Substituting the values of $\frac{d\beta}{d\theta}$ and $\frac{d^2\beta}{d\theta^2}$ in (24) we have

$$\frac{d^2x_1}{d\theta^2} = R\left[\cos\theta + \frac{n-l+k}{n}\left\{\frac{n^2\cos^2\theta}{[n^2-(\sin\theta-b)^2]^{\frac{3}{2}}} - \frac{\sin\theta(\sin\theta-b)}{\sqrt{n^2-(\sin\theta-b)^2}}\right\} + \frac{c\sin\theta}{n}\right],$$

and on introducing the value of Z, given in (5), this becomes

$$\frac{d^2x_1}{d\theta^2} = R\left\{\cos\theta + \frac{n-l+k}{n}Z + \frac{c\sin\theta}{n}\right\}.$$

Let F_{dx} be the force required to produce this acceleration, and we have

$$F_{dx} = \frac{1}{2}m\tau^2 R\left\{\cos\theta + \frac{n-l+k}{n}Z + \frac{c\sin\theta}{n}\right\}. \tag{27}$$

Dividing this between the crank- and wrist-pins gives for the portions acting at the crank-pin A

$$F_{dx}A = \frac{1}{2}m\tau^2 R\left[\frac{l-k+c\cot\beta}{n}\left\{\cos\theta + \frac{n-l+k}{n}Z + \frac{c\sin}{n}\right\}\right]. \tag{28}$$

The force required to produce the X acceleration of the mass E, obtained in a similar way to F_{dx}, is

$$F_{ex} = \frac{1}{2}m\tau^2 R\left\{\cos\theta + \frac{n-l-k}{n}Z + \frac{c\sin\theta}{n}\right\}, \tag{29}$$

and its component acting at the crank-pin will be

$$F_{ex}A = \frac{1}{2}m\tau^2 R\left[\frac{l+k+c\cot\beta}{n}\left\{\cos\theta + \frac{n-l-k}{n}Z + \frac{c\sin\theta}{n}\right\}\right]. \tag{30}$$

Adding these two components as given in equations (28) and (30), we have the total accelerating force X_2 at the crank-pin, or

$$F_{dx}A + F_{ex}A = X_2;$$

from which

$$X_2 = m\tau^2 R\left[\left(\frac{l}{n} + \frac{c\sqrt{n^2 - (\sin\theta - b)^2}}{n(\sin\theta - b)}\right)\left(\cos\theta + \frac{n-l}{n}Z + \frac{c\sin\theta}{n}\right) - \frac{k^2}{n^2}Z\right]. \quad (31)$$

The accelerating force X_1 acting at the wrist-pin may be found by obtaining the sum of the forces in the X direction at the masses D and E and subtracting X_2, or

$$X_1 = F_{dx} + F_{ex} - X_2;$$

from which

$$\begin{aligned} X_1 &= m\tau^2 R\left(\cos\theta + \frac{n-l}{n}Z + \frac{c}{n}\sin\theta\right) - X_2 \\ &= m\tau^2 R\left[\left(\frac{n-l}{n} - \frac{c\sqrt{n^2 - (\sin\theta - b)^2}}{n(\sin\theta - b)}\right)\left(\cos\theta + \frac{n-l}{n}Z + \frac{c\sin\theta}{n}\right) + \frac{k^2}{n^2}Z\right]. \end{aligned} \quad (32)$$

To determine the accelerating forces in the direction of the axis of Y we have, for the mass at D,

$$y_1 = R\,[(l-k)\sin\beta + c\cos\beta];$$

from which

$$\frac{d^2y_1}{d\theta^2} = -R\left(\frac{l-k}{n}\sin\theta + \frac{c}{n}Z\right). \quad (33)$$

Similarly, for the mass at E

$$\frac{d^2y_2}{d\theta^2} = -R\left(\frac{l+k}{n}\sin\theta + \frac{c}{n}Z\right). \quad (34)$$

If F_{dy} and F_{ey} be the Y accelerating forces for the masses D and E respectively, we have from equations (33) and (34)

$$F_{dy} = -\frac{1}{2}m\tau^2 R\left(\frac{l-k}{n}\sin\theta + \frac{c}{n}Z\right), \quad (35)$$

and

$$F_{ey} = -\frac{1}{2}m\tau^2 R\left(\frac{l+k}{n}\sin\theta + \frac{c}{n}Z\right). \quad (36)$$

The component of F_{dy} acting at the crank-pin will be

$$F_{dy}A = \frac{l - k - c \tan \beta}{n} F_{dy}$$

$$= -\frac{1}{2} m\tau^2 R\left[\frac{l - k - c \tan \beta}{n}\left(\frac{l-k}{n} \sin \theta + \frac{c}{n} Z\right)\right]; \tag{37}$$

and of F_{ey}

$$F_{ey}A = \frac{l + k - c \tan \beta}{n} F_{ey}$$

$$= -\frac{1}{2} m\tau^2 R\left[\frac{l + k - c \tan \beta}{n}\left(\frac{l+k}{n} \sin \theta + \frac{c}{n} Z\right)\right]. \tag{38}$$

The Y component of the accelerating force acting at the crank-pin will be equal to

$$F_{dy}A + F_{ey}A$$

or

$$Y_2 = F_{dy}A + F_{ey}A$$

$$= -m\tau^2 R\left[\left(\frac{l}{n} - \frac{c\,(\sin \theta - b)}{n\sqrt{n^2 - (\sin \theta - b)^2}}\right)\left(\frac{l}{n} \sin \theta + \frac{c}{n} Z\right) + \frac{k^2}{n^2} \sin \theta\right]. \tag{39}$$

The accelerating force Y_1 acting at the wrist-pin may be found by taking the sum of the forces in the Y direction that accelerate the masses D and E as given in (35) and (36) and subtracting Y_2, or

$$Y_1 = F_{dy} + F_{ey} - Y_2$$

$$= -m\tau^2 R\left(\frac{l}{n} \sin \theta + \frac{c}{n} Z\right) - Y_2$$

$$= -m\tau^2 R\left[\left(\frac{n-l}{n} + \frac{c\,(\sin \theta - b)}{n\sqrt{n^2 - (\sin \theta - b)^2}}\right)\left(\frac{l}{n} \sin \theta + \frac{c}{n} Z\right) - \frac{k^2}{n^2} \sin \theta\right]. \tag{40}$$

Equations (40), (39), (32), and (31) are the same as (20), (21), (22), and (23); the first and second methods therefore give the same results.

RATIO BETWEEN THE LENGTH OF STROKE OF THE PISTON AND THE RADIUS OF THE CRANK.

The length of stroke s will be the distance travelled by the wrist-pin between the two positions at which $\theta = -\beta$,

or $$s = R\sqrt{(n+1)^2 - b^2} - R\sqrt{(n-1)^2 - b^2}. \tag{41}$$

DETERMINATION OF THE NET FORCES ACTING AT VARIOUS POINTS.

The forces concerned in the action of an engine are the force of the steam on the piston, the forces required to accelerate the parts, and their weight and friction. We will first determine the effect of the pressure of the steam combined with the accelerating forces, omitting the weight and friction of the parts.

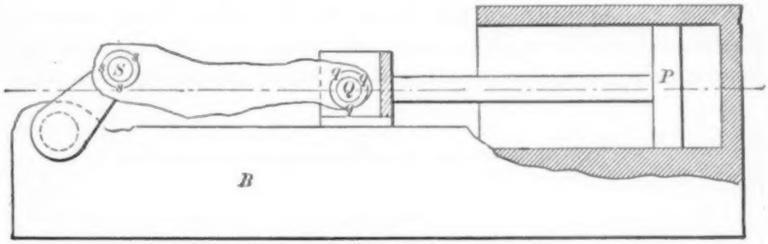


FIG. 253.

Let P_a = effort of the steam on the piston;
G = reaction of cross-head guides;
P_w = force exerted by the connecting-rod upon the wrist-pin;
P_c = force exerted by the connecting-rod upon the crank-pin;
R_1 = resultant in direction of the centre line of the rod;*
T = tangential component of P_c;
N = radial component of P_c;
H_c = force exerted by the bed on the crank-shaft.

The force exerted by the steam on the piston is equal to the difference of the forces exerted on the two cylinder heads, or to the force acting on the cylinder as a whole; its equilibrium may, therefore, be represented by two lines representing the pressure in magnitude and direction and directly opposed to each other, as shown in Fig. 254, the resultant of which will be zero.

* In our analysis we have determined the forces at the wrist and crank-pins that will be equivalent in accelerating effect to the summation of the elementary accelerating forces acting at all portions of the mass of the rod, but these components cannot be used to obtain the internal strains at each portion of the rod, unless the law of the forces acting at the separate elements of its mass is included in the problem. R_1 does not, therefore, represent the actual force along the line of centres of the rod, but simply the resultant in that direction that must be common to the polygons of equilibrium at each of the pins. R_1 will have a variety of values, according to the choice of the forces to give the accelerating moment; but it is a principle in mechanics that the external forces cannot be altered by this means.

The equilibrium of the piston, piston-rod, and cross-head (see P, Fig. 253) is represented in Fig. 255.

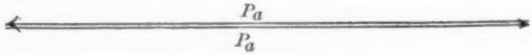


FIG. 254.

From the pressure of the steam on the piston there must be subtracted the force F_1 required to accelerate the mass of the piston, piston-rod, and cross-head, in order to determine the force

FIG. 255.

in the direction of the motion of the piston that acts on the wrist-pin; let this force be F, and it follows that

$$F = P_a - F_1. \tag{42}$$

The equilibrium of the wrist-pin (see Q, Fig. 253) is represented in Fig. 256. The forces transmitted from the cross-head to the wrist-pin are the horizontal force F and the reaction of the guides G, and these are in equilibrium with the force P_w.

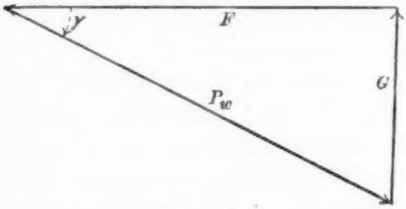


FIG. 256.

From Fig. 256 it follows that

$$P_w = \sqrt{(P_a - F_1)^2 + G^2} \tag{43}$$

Let the angle made by P_w with the axis of X be γ; then

$$P_w = F \sec \gamma = (P_a - F_1) \sec \gamma. \tag{44}$$

The equilibrium of the wrist-pin box (see q, Fig. 253) is represented in Fig. 257. The force P_w is in equilibrium with the force R_1, in the direction of the line of centres of the connecting rod and the accelerating force at the wrist-pin. The accelerating force at the wrist-pin is the resultant of its two components X_1

and Y_1, and these components, and not the force itself, are represented in the figure.

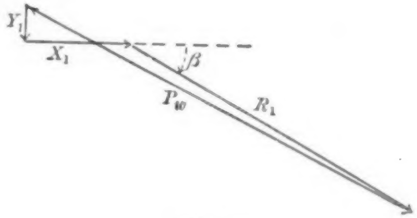


FIG. 257.

From Fig. 257 we have

$$R_1 = (P_w \cos \gamma - X_1) \sec \beta. \tag{45}$$

The equilibrium of the crank-pin box (see s, Fig. 253) is represented in Fig. 258. X_2 and Y_2, together with R_1, are in equilibrium with the pressure P_c. From the figure we have

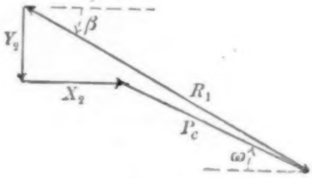


FIG. 258.

$$P_c = \sqrt{[(R_1 \cos \beta - X_2)^2 + (R_1 \sin \beta - Y_2)^2]} \tag{46}$$
$$= \sqrt{\{(P_a - F_1 - X_1 - X_2)^2 + [(P_a - F_1 - X_1) \tan \beta - Y_2]^2\}}.$$

Let the angle made by P_c with the axis of X be ω; then

$$P_c \cos \omega = R_1 \cos \beta - X_2. \tag{47}$$

The equilibrium of the crank pin (see S, Fig. 253) is represented in Fig. 259.

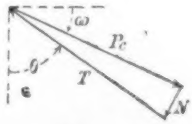


FIG. 259.

From Fig. 259 we have

$$T = P_c \cos (90 - \theta + \omega)$$
$$= P_c (\sin \theta \cos \omega + \cos \theta \sin \omega).$$

From Fig. 258

$$P_c \sin \omega = R_1 \sin \beta - Y_2. \tag{48}$$

On introducing the values given in (47), (48), (45), and (44), in the above equation for T, we have

$$T = (P_a - F_1 - X_1 - X_2) \sin \theta - Y_2 \cos \theta + (P_a - F_1 - X_1) \tan \beta \cos \theta;$$

from which

$$T = (P_a - F_1 - X_1) \sec \beta \sin (\theta + \beta) - Y_2 \cos \theta - X_2 \sin \theta. \quad (49)$$

We may obtain, in a similar manner,

$$N = (P_a - F - X_1) \sec \beta \cos (\theta + \beta) + Y_2 \sin \theta - X_2 \cos \theta. \quad (50)$$

It also follows from Fig. 259 that

$$P_c = \sqrt{T^2 + N^2}. \quad (51)$$

The equilibrium of the crank shaft is represented in Fig. 260. The force P_c acting on the crank-pin will be transmitted to the crank-shaft, and this force will be in equilibrium with the pressure on the crank-shaft bearings H_c, and the weight of the fly-wheel, and the force exerted on the shaft by the mechanism that transmits the power developed by the engine, the sum of which we will call H_1.

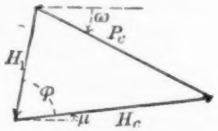


FIG. 260.

Let φ be the angle made by H_1 with the axis of X, and μ that made by H_c; then from Fig. 260

$$H_c \cos \mu = P_c \cos \omega + H_1 \cos \varphi.$$

Introducing values given in (48) and (45), this becomes

$$H_c = (P_a - F_1 - X_1 - X_2) \sec \mu = H_1 \frac{\cos \varphi}{\cos \mu}. \quad (52)$$

The equilibrium of the bed of the engine (see B, Fig. 253) is represented in Fig. 261.

The forces acting on the bed are the force P_a at the cylinder head, the reaction of the guides G, the force H_c at the crank-shaft journals, and the weight of the bed, which we will call B_w, acting at the angle δ with the axis of Y. These will be in equilibrium with the force between the foundation and the bed of the

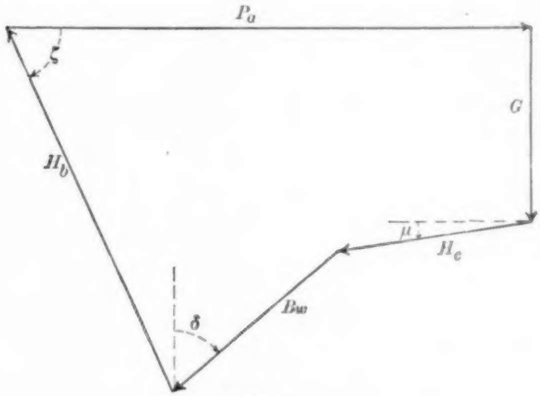


FIG. 261.

engine. Let this force be H_b, and the angle it makes with the axis of X be ζ; then it follows from Fig. 261 that

$$H_b \cos \zeta = P_a - H_c \cos \mu - B_w \sin \delta;$$

from which

$$H_b = (P_a - H_c \cos \mu - B_w \sin \delta) \sec \zeta. \tag{53}$$

The following value of H_b may also be obtained:

$$H_b = \sqrt{[(F_1 + X_1 + X_2 - H_1 \cos \varphi - B_w \sin \delta)^2 + (B_w \cos \delta + G + {}_c H \sin \mu)^2]}. \tag{54}$$

From Fig. 260 we have

$$H_c \sin \mu = H_1 \sin \varphi - P_c \sin \omega,$$

which substituted in equation (54) gives

$$H_b = \sqrt{[F_1 + X_1 + X_2 - H_1 \cos \varphi - B_w \sin \delta)^2 + (B_w \cos \delta + G + H_1 \sin \varphi - P_c \sin \omega)^2]}$$

From Figs. 255, 257, and 258 we obtain

$$P_c \sin \omega = G - Y_1 - Y_2,$$

which reduces the above value of H_b to

$$H_b = \sqrt{[(F_1 + X_1 + X_2 - H_1 \cos \varphi - B_w \sin \delta)^2 + (H_1 \sin \varphi + B_w \cos \delta + Y_1 + Y_2)^2]}. \tag{55}$$

The X component of H_b, which we will call H_{bx}, is

$$H_{bx} = H_b \cos \zeta.$$

Substituting the value of H_b as given in (53), this becomes

$$H_{bx} = F_1 + X_1 + X_2 - H_1 \cos \varphi - B_w \sin \delta.$$

It follows from (55) that the Y component will be

$$H_{by} = Y_1 + Y_2 + H_1 \sin \varphi + B_w \cos \delta.$$

If X_t and Y_t are the components of the translative force tending to shake the bed, they will act in an opposite direction to the forces that act from the foundation to the bed. As a constant force does not tend to produce a shake, X_t and Y_t will only contain that portion of H_{bx} and H_{by} that is variable. The weight of the foundation B_w will be constant, and, provided the method of transmitting power from the engine does not alter the value of H_1, which is ordinarily very nearly so, its value may also be considered as constant, and we will have

$$X_t = -F_1 - X_1 - X_2, \tag{56}$$

and

$$Y_t = -Y_1 - Y_2. \tag{57}$$

These forces, as has already been stated, tend to translate the bed of the engine, and do not indicate the value of the moment that may be produced by forces opposite in direction and not having the same line of action.

Figs. 254-261 may be combined, and a polygon formed that represents the equilibrium of the entire engine, as shown in Fig. 262.

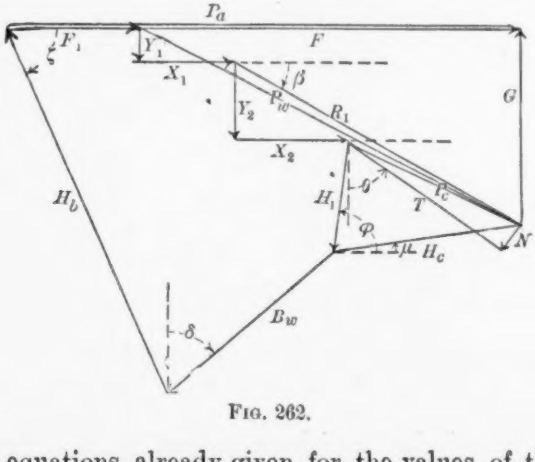


FIG. 262.

All the equations already given for the values of the forces may be derived from Fig. 262.

Substituting the value of H_b as given in (53), this becomes

$$H_{bx} = F_1 + X_1 + X_2 - H_1 \cos \varphi - B_w \sin \delta.$$

It follows from (55) that the Y component will be

$$H_{by} = Y_1 + Y_2 + H_1 \sin \varphi + B_w \cos \delta.$$

If X_t and Y_t are the components of the translative force tending to shake the bed, they will act in an opposite direction to the forces that act from the foundation to the bed. As a constant force does not tend to produce a shake, X_t and Y_t will only contain that portion of H_{bx} and H_{by} that is variable. The weight of the foundation B_w will be constant, and, provided the method of transmitting power from the engine does not alter the value of H_1, which is ordinarily very nearly so, its value may also be considered as constant, and we will have

$$X_t = -F_1 - X_1 - X_2, \tag{56}$$

and

$$Y_t = -Y_1 - Y_2. \tag{57}$$

These forces, as has already been stated, tend to translate the bed of the engine, and do not indicate the value of the moment that may be produced by forces opposite in direction and not having the same line of action.

Figs. 254–261 may be combined, and a polygon formed that represents the equilibrium of the entire engine, as shown in Fig. 262.

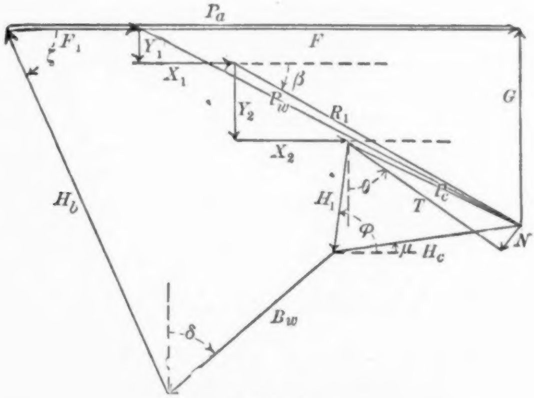


FIG. 262.

All the equations already given for the values of the forces may be derived from Fig. 262.

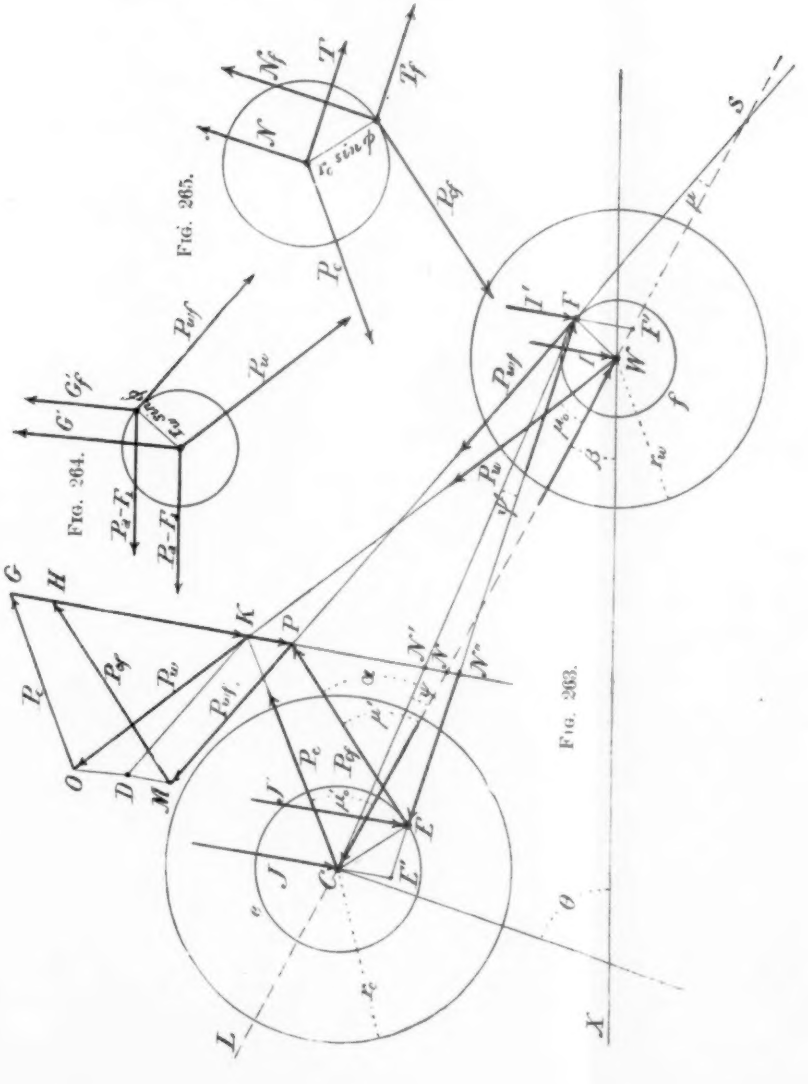


FIG. 263.

FIG. 264.

FIG. 265.

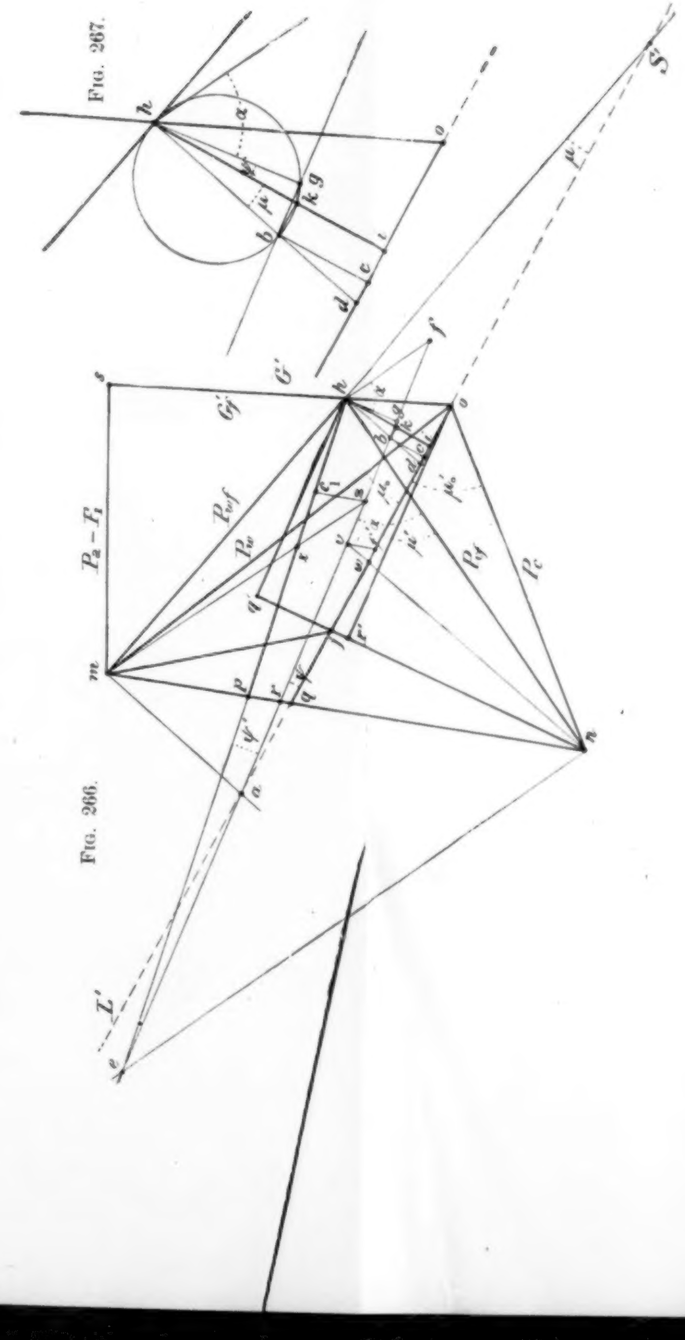


FIG. 266.

FIG. 267.

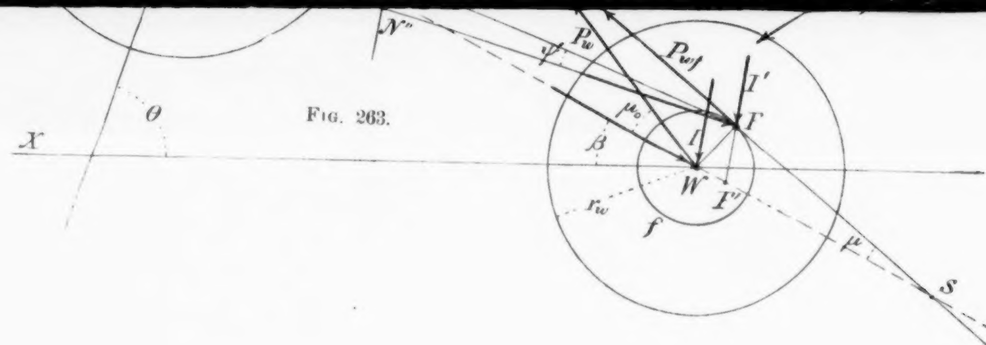


FIG. 263.

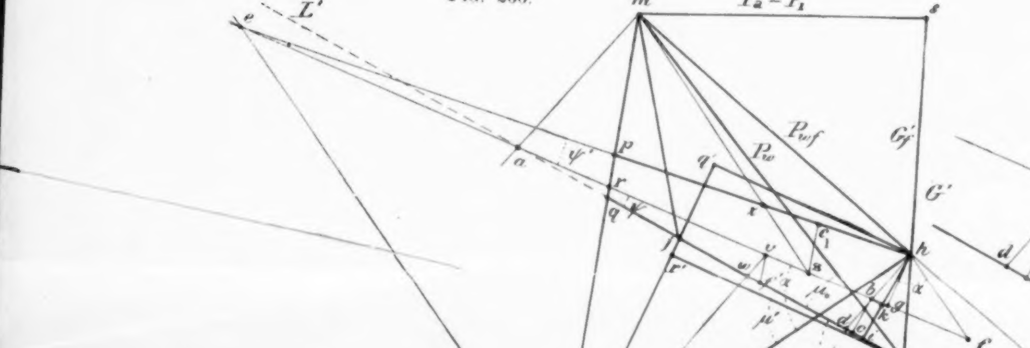


FIG. 266.

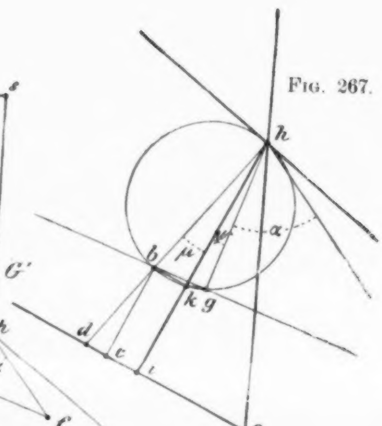
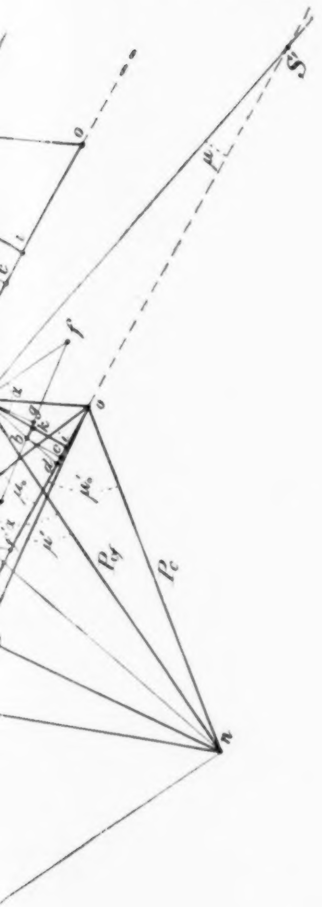


FIG. 267.

D. S. JACOBUS.



Equation (49) may be written

$$T = \left[P_a - \left[F_1 + X_1 + \frac{Y_2 \cos\theta + X_2 \sin\theta}{\sec\beta \sin(\theta + \beta)} \right] \right] \sec\beta \sin(\theta + \beta).$$

The expression

$$F_1 + X_1 + \frac{Y_2 \cos\theta + X_2 \sin\theta}{\sec\beta \sin(\theta + \beta)},$$

which we will call P_b, therefore represents a pressure which, on being subtracted from P_a, will give a force that may be resolved as if no motion were present in order to obtain T, or

$$T = (P_a - P_b) \sec\beta \sin(\theta + \beta). \tag{58}$$

P_b may be laid off on the indicator diagram, and its value thus subtracted directly from the steam pressure.

FRICTION AND GRAVITY INCLUDED.

It has been demonstrated, in a paper prepared by Professor Webb and the writer,* that the effect of the friction at the connecting-rod bearings may be determined by introdu ing into the equilibrium polygons two forces, A and B, at right angles to the centre line of the rod, the sum of A and B being applied at each of the pins in opposite directions. The values of A and B are

$$A = \frac{P_{wf} r_w \sin\varphi_w}{nR}, \text{ and } B = \frac{P_{cf} r_c \sin\varphi_c}{nR},$$

in which r_w and r_c are equal, respectively, to the radii of the wrist and crank, and sin φ_w and sin φ_c to the sines of the angles of friction. The effect of the weight was also included, and equilibrium polygons given showing the relation of the forces at the wrist- and crank-pins, together with equations deduced from the same.

* *Annals of Mathematics*, December, 1888.

EFFECT OF FRICTION AT CONNECTING-ROD BEARINGS ON THE FORCES TRANSMITTED.*

BY PROFS. J. BURKITT WEBB † AND D. S. JACOBUS, HOBOKEN, N. J.

Fig. 263 represents the equilibrium of the forces acting on a connecting rod, with a centre-line WC, and wrist- and crank-pins indicated by circles about W and C whose radii are respectively r_w and r_c.

Figs. 264 and 265 represent the equilibrium of the forces acting at the wrist- and crank-pins.

The forces shown in these figures are as follows:

$P_a - F_1$ = pressure of steam less the accelerating force for piston, piston-rod, and cross-head;

G' = the pressure of the cross-head guides against the cross-head for frictionless pins, acting at the centre of the wrist-pin at the angle $90° + \varphi'$. This angle is the same for frictionless or rough pins; ‡

G'_f = the pressure when there is friction.

P_c and P_w = pressures of the pins C and W against the rod for frictionless pins;

P_{cf} and P_{wf} = the same when there is friction;

$KG = PH$ = the single force which, if applied in the line NG, will support the weight and correctly accelerate the rod;

I and J = pressures of the ends of the rod at the pin-centres W and C due to its acceleration and weight and for frictionless pins;

I' and J' = the same pressures for rough pins acting at the points F and E of the pins, F being the foot of a perpendicular from W on P_{wf} and E the foot of a perpendicular from C on P_{cf}.

The moments of I and I' about N must evidently be equal and opposite to those of J and J' about the same point; from which it follows that

$$I : J = CN : NW$$

and

$$I' : J' = EN'' : N''F.$$

---

* Reprinted from the *Annals of Mathematics*, December, 1888.

† Professor Jacobus insists upon my name appearing first in this article. I fully appreciate the courtesy, but it is hardly fair to himself, as he has done most of the work. J. B. W.

‡ φ' = angle of friction at cross-head guides.

In addition to the above, the following lines will be needed for the purpose of demonstration:

Ee and *Ff* friction circles about C and W, the radii of which are respectively equal to $r_c \sin \varphi$ and $r_w \sin_t \varphi$;

FE and *FC* connecting F with E and C;

MD from M, parallel and equal to *PK* or *GH*;

DO connecting D with O.

Fig. 266 is composed of various equilibrium polygons representing the equilibrium of the forces acting on various parts, each polygon being drawn with and without friction; these forces are, of course, equal and parallel to those in Fig. 263, *nm* being the force due to the weight and acceleration, and therefore the same as *KG* and *PH*. Fig. 267 is an enlargement, for the sake of clearness, of a portion of Fig. 266.

smo is the equilibrium triangle of the forces acting on the wrist-pin when there is no friction on wrist- or crank-pin.

mno is the equilibrium triangle of the forces acting on the rod, as a whole, for frictionless pins, and *mnh* the same when there is friction. The introduction of the force *oq* transmitted by the rod from wrist- to crank-pin divides *mn* into *mq* and *qn*, forces applied at the wrist- and crank-pins, thereby cutting *mno* into two polygons, *mqo* representing the equilibrium of the forces acting on the crank-pin and *oqn* representing forces on the wrist-pin. In the same way, *hp* divides the polygon *mnh* into *mph* and *hpm*.

mjo and *njo*, and *mhiq* and *qihn*, are also polygons for wrist- and crank-pins.*

In addition to the above, the following lines will be needed for the purpose of demonstration:

ma perpendicular to P_{wf};

af parallel to *CF* in Fig. 263.

hf perpendicular to P_{cf};

ne perpendicular to P_{cf};

---

* *mq* and *qn* are the simplest set of forces that will support the weight and produce the required acceleration, inasmuch as they cause no unnecessary tension in the rod. Any pair of forces having *mn* for a resultant will give the support and acceleration; consequently we may make various convenient suppositions in regard to these forces, as, for instance, we may suppose that the crank-pin force remains normal to the crank-circle, which gives us *mj* and *jn*. The intersection of any such pair of forces will fall on the line *LS*, and any pair may be changed to another by introducing two equal and opposite forces at the ends of the rod; thus, *qj* combined with *mq* produces *mj*, and *jq* combined with *qn* produces *jn*.

eh parallel to *EF* (Fig. 263) through *e*, the intersection of *ne* and *af*, and passing therefore through *h*, as will be shown;

mxz perpendicular to P_{cf} through *m*;

zc_1 parallel to *mn* through *z*, the intersection of *mx* with *ef*;

nwv perpendicular to P_{wf} through *n*;

vf' parallel to *mn* through *v*, the intersection of *nw* with *ef*;

hd perpendicular to P_{wf};

hi perpendicular from *h* on $L'S'$, which is parallel to the centre-line *LS* (Fig. 263).

bc perpendicular on $L'S'$ from *b*, the intersection of *hd* with *af*;

hg perpendicular from *h* on *ef*;

bk perpendicular from *b* on *hi*;

bkgh a semicircle containing the right angles *bkh* and *bgh*;

kg a line completing the triangle *ghk*, which is similar to *dab*, similar to *WCF* (Fig. 263), as will be proved;

nq' through *n* parallel to the crank radius;

mj connecting *m* with *j*, the intersection of nq' with *ao*;

or perpendicular from *o* on nq';

hq' perpendicular from *h* on nq';

Fig. 266 contains the following forces:

$sm = P_a - F_1$;

om and $hm = P_w$ and P_{wf},

on and $hn = P_c$ and P_{cf};

os and $hs = G$ and G'_f, the guide reactions;

$mn =$ the resultant of the weight of the rod at its centre of gravity and the forces due to the acceleration of the different parts of the rod. *nm* is therefore that single force which, if applied at the proper point, would support and correctly accelerate the rod; *nm* is also the resultant of such parts of the wrist- and crank-pin pressures as give the support and acceleration;

mp and *mq* and *pn* and $qn =$ the parts of the wrist- and crank-pin pressures which actually give the support and acceleration;

hp and $oq =$ the force transmitted from wrist- to crank-pin by the rod, with and without friction, *hp* obliquely from *F* to *E*, *oq* axially;

$oh =$ change in guide reaction due to pin friction;

$$bc \text{ and } hk = A \text{ and } B = \frac{P_{wf}r_w \sin\varphi}{nR} \text{ and } \frac{P_{cf}r_c \sin\varphi}{nR} \text{ and}$$

$hi = bc + hk = A + B$, two forces which will be explained in the proper place;

$jo =$ that portion of the force transmitted from the wrist- to

the crank-pin that performs work upon the fly-wheel, for frictionless pins;

qi = the force transmitted axially if P_{wf} and P_{cf} be supposed to act at the centres of the pins, and moments be introduced to counterbalance the effect of altering their points of application;

or' = force acting on crank-pin in tangential direction for frictionless pins;

hq' = the same when there is friction.

We will suppose the force $P_a - F_1$ applied at the wrist-pin W to be known, and proceed to determine the force that reaches the crank-pin C.

For frictionless pins we proceed as follows:

Construct the polygon $smqo$ by making sm and mq equal and parallel, respectively, to $P_a - F_1$ and I, and drawing from q and s lines parallel to CW and G', marking their intersection o; then will os be the guide reaction and oq the force transmitted from W to C; om will be P_w the resultant of the forces G' and $P_a - F_1$. Next lay off qn equal and parallel to J and complete the triangle mno; then will $no = P_c$, the pressure of the crank-pin upon the connecting-rod. This pressure P_c is pressure available for doing work upon the crank and fly-wheel.

If there is friction the construction of the diagrams is more complicated, for the introduction of friction not only changes the direction and magnitude of the forces, but also alters their points of application.

There are three conditions that govern the construction of a diagram including the effect of friction; these are:

(a) The resultant of P_{cf} and P_{wf} must equal in magnitude and direction, and have the same line of action, NG, as the resultant of P_c and P_w;

(b) The projections of P_{wf} and P_w on a line perpendicular to the guide reaction must be equal to each other and to that of $P_a - F_1$;

(c) The presence of friction changes the forces P_c and P_w into P_{cf} and P_{wf}, which no longer pass through the centres of the pins, but are tangent to circles Ee and Ff, whose radii are constant and respectively equal to $r_c \sin \varphi$ and $r_w \sin \varphi$.

The last of these conditions is generally given by authorities, such as Rankine and Weisbach, and requires therefore no demonstration.*

* See Rankine's *Machinery and Mill Work*, p. 428.

That the resultant of P_{cf} and P_{wf} should be the same as that of P_c and P_w, arises from the fact that either is the single force PH or KG which would support the weight and produce the required acceleration of the rod, if applied at the proper point, and must therefore be independent of the friction of the pins.

The second of these conditions is evident from Fig. 266, where *mos* is the triangle of forces acting on a frictionless wrist-pin, and *mhs* the triangle with friction considered: obviously P_w and P_{wf} have the same component perpendicular to *os* as $P_a - F_1$ has.

If we attempt to apply these principles directly to Fig. 264, they lead, respectively, to the following geometrical conditions:

(*a*) HP must be equal in length and direction to GK, and must lie in the same line GN;

(*b*) DO is parallel to G';

(*c*) PF is perpendicular to $FW = r_w \sin \varphi$, and PE is perpendicular to $EC = r_c \sin \varphi$.

The construction of the diagram under these conditions can only be accomplished by approximate methods more difficult of application than those which we will explain in the construction of Fig. 266.

The geometrical conditions for the construction of Fig. 266 are as follows:

(*a*) *mn* is common to both the polygons *mnos* and *mnhs*;

(*b*) *ohs* is a right line;

(*c*) In applying the third principle we meet with the only difficulty in the construction of Fig. 266; a fundamental principle in mechanics, however, furnishes us with a simple means of solving the problem with all the accuracy desirable.

Inasmuch as the effect of friction is to displace the forces which the pins exert upon the rod, so that they become tangent to the friction circles previously mentioned, these forces will exert moments upon the rod tending to rotate it, and therefore affecting the other forces of the system, so that the forces P_c and P_w become P_{cf} and P_{wf}, besides becoming displaced from C to E and W to F. By a principle in mechanics a force applied at any point, as F, is equivalent to an equal force, in magnitude and direction, at any other point, as W, plus a moment equal to the force multiplied by the perpendicular distance through which it has been displaced. We may, therefore, suppose the forces P_{cf} and P_{wf} to be applied at the centres of the pins if, at the same time, we introduce the moments

$$P_{wf} r_w \sin \varphi \text{ and } P_{cf} r_c \sin \varphi.$$

For convenience we will suppose each moment to be produced by a pair of equal and opposite forces acting perpendicular to the rod at W and C. Calling these forces A and B, and letting the distance $WC = nR = n \times$ crank radius, we shall have

$$AnR = P_{wf} r_w \sin \varphi,$$

and
$$BnR = P_{cf} r_c \sin \varphi,$$

so that the moment to be introduced will be

$$AnR + BnR = (A + B) nR.$$

According to this, instead of supposing P_{wf} and P_{cf} to act on the rod at F and E, no error will be involved if we suppose them to act at W and C and introduce two additional forces, $A + B$ acting at W perpendicular to WC, and $-(A + B)$ acting at C. A and B appear also in the equilibrium polygons, Figs. 268 and 269.*

The important advantage of this change is that it allows the use of the known accelerating forces I and J in place of the unknowns I' and J'.

The construction of Fig. 266 by the aid of $A + B$ is as follows:

Having laid off $P_a - F_1$ and mq as before, and drawn through q and s lines qo and so, respectively parallel to the rod and the guide reaction, cut so by a line parallel to qo and at a distance from it $= A + B$. This will give the point h, from which the perpendicular $hi = A + B$ may be let fall upon qo. The polygon $smqih$ will now represent the equilibrium of the forces acting on the wrist-pin, and hs will be the new value of the guide reaction. Connecting h with m and n we obtain P_{wf} and P_{cf}.

The value of $A + B$ can be obtained with an exactness amply sufficient by using the values of P_w and P_c in place of those of P_{wf} and P_{cf}, as will be shown in the case of a horizontal high-speed engine, to which formulæ derived by this method have been applied. This form of the result has the advantage that it enables the same to be easily applied.

As the above reasoning may seem to involve a departure from

* When the angle β is increasing A is $+$. B is $+$ or $-$ according to whether θ is supposed to increase or diminish.

the exact conditions of the problem, in which the forces are really applied at F and E, we will proceed to prove that the introduction of A and B into the solution for frictionless pins will give the correct result when friction is considered, that is:

To prove that $hi = A + B$.

This proof divides itself into three parts, as follows:

(*a*) To prove that $cb = A$;

(*b*) To prove that $kh = B$;

(*c*) To prove that $ih = kh + cb$;

(*a*) From the similar triangles CFW and abd we have

$$\frac{WF}{CW} = \frac{db}{ad};$$

but

$$ad = \frac{P_{wf}}{\cos \mu},$$

because am and dh are perpendicular to mh, and ad makes with mh the angle μ, also

$$CW = nR,$$

and

$$WF = r_w \sin \varphi;$$

substituting these values, we have

$$db = \frac{P_{wf} r_w \sin \varphi}{nR \cos \mu};$$

but

$$cb = db \cos \mu;$$

$$\therefore cb = \frac{P_{wf}\, r_w \sin \varphi}{nR} = A.$$

(*b*) To prove that $kh = B$.

To prove this we must first show the following:

I. That a line through e parallel to EF divides mn at p into the two forces $mp = I'$, and $pn = J'$ acting at F and E.

II. That the line ep produced will pass through h.

III. That hgk is similar to CWF.

I. To show that mn is correctly divided at p; that is, to show that

$$mp = I', \text{ and } pn = J'.$$

The relations previously given between I, I', J, and J' are

$$I + J = I' + J' = mn,$$

$$I : J = CN : NW,$$

and $$I' : J' = EN'' : N''F.$$

By construction, therefore, $mn = I' + J'$, and we have only to prove that

$$mp : pn = I' : J'.$$

First, to prove that

$$mr : nr = CN : NF' :$$

q being the point of division for frictionless pins, we know that

$$mq : qn = CN : NW,$$

and also, from the similar triangles amq and wnq, we have

$$mq : qn = aq : qw;$$

consequently $$aq : qw = CN : NW.$$

This, in addition to the similarity of the triangles CFW with avw, CFF' with avf', and $CN'N$ with arq, gives us two similar figures contained in Figs. 263 and 266; viz.,

$$CN'FF'WNC$$

similar to $$arvf'wqa,$$

from which it follows that

$$CN : NF' = aq : qf'.$$

But $$aq : qf' = ar : rv,$$

because arq and avf' are similar, and by the similar triangles mra and nrv we have

$$ar : rv = mr : nr;$$

$$\therefore CN : NF' = mr : nr,$$

which was to be proved.

Second, to prove that

$$mp : pn = EN'' : N''F :$$

From the similar triangles ner and mzr we have

$$mr : rn = zr : re;$$

but $$mr : rn = CN : NF'$$

$$= CN' : N'F;$$

$$\therefore zr : re = CN' : N'F.$$

This, in addition to the similarity of the triangles $FE'C$ with ec_1z, FEC with exz_1, and CEE' with zxc, gives us two similar figures contained in Figs. 263 and 266; viz.,

$$FN'CE'EN''F$$

similar to $erzc_1xpe$,
from which it follows that

$$xp : pe = EN'' : N''F;$$

From the similar triangles mxp and nep we have

$$mp : pn = xp : pe;$$

$$\therefore mp : pn = EN'' : N''F;$$

but
$$EN'' : N''F = I' : J';$$

$$\therefore mp : pn = I' : J',$$

which was to be proved.

II. To show that ep passes through h.

At F the force P_{wf} of the pin against the rod is divided into I' and the force transmitted from F to E, or, in other words P_{wf} is the resultant of the latter forces, and when reversed will form with them a triangle of forces. This triangle mhp appears in Fig. 266, in which $pm = I'$ and $mh = -P_{wf}$, and the third side, having been drawn through p and parallel to FE, must pass through h, so as to make hp the force transmitted from F to E.

III. To show that hgk is similar to CWF.

The angle kgh is equal to $hgb + bgk = 90° + \mu$, because hgb is inscribed in the semicircle bgh, and $bgk = bhk = \mu$; also, khg is equal to $fao = \psi$, because gh and kh are perpendicular to fa and ox; therefore the triangle hkg has two of its angles the same as the angles of CWF, in which $FCW = \psi$ and $CWF = 90° + \mu$, it being the exterior angle of the right-angle triangle WFS.

We are now prepared to prove that $kh = B$. From the similar triangles FEC and ehf we have

$$\frac{hf}{ef} = \frac{CE}{CF};$$

but
$$CE = r_c \sin \varphi,$$

and $$ef = \frac{P_{ef}}{\cos\alpha};$$

$$\therefore hf = \frac{P_{ef}r_c \sin\varphi}{CF\cos\alpha};$$

but $$gh = hf\cos\alpha;$$

$$\therefore gh = \frac{P_{ef}r_c \sin\varphi}{CF}.$$

From the similar triangles hgk and CWF we have

$$\frac{gh}{kh} = \frac{CW}{CF} = \frac{nR}{CF};$$

$$\therefore kh = gh\frac{CF}{nR}.$$

Substituting the value of gh and reducing, we have

$$kh = \frac{P_{ef}r_c \sin\varphi}{nR} = B.$$

as was to be proved.

(c) To prove that $ih = kh + cb$.

This follows because bk is perpendicular to hi, and therefore $bc = ki$; consequently, as

$$cb = A \text{ and } kh = B,$$

$$ih = A + B,$$

as was to be proved.

ANALYTICAL EXPRESSION OF THE FOREGOING PRINCIPLES.

Let R = radius of crank circle;

nR = length of connecting-rod;

lR = distance from the wrist-pin to the foot of the perpendicular let fall from the centre of gravity of the rod upon its centre-line;

cR = distance of the centre of gravity from the centre-line of the rod;

θ and β = angles of crank and connecting-rod with the path of the wrist-pin;

δ = angle made by tipping the engine up about the crank-shaft;

$\tan \varphi$ = coefficient of friction at wrist- and crank-pins;

$\tan \varphi'$ = coefficient of friction at cross-head guides;

τ = angular velocity of crank;

W_1 and M = weight and mass of the piston, piston-rod, and cross-head;

W_2 and m = weight and mass of the connecting-rod;

C = component of the weight of the piston, piston-rod, and cross-head that acts in the direction of the centre-line of the cylinder.

D and E = portions of the weight of the connecting-rod borne respectively by the wrist- and crank-pins;

H = friction of the piston and piston-rod;

X_1 and X_2 = components, in the direction of the line of travel of the wrist-pin, of the accelerating forces at the wrist- and crank-pins;

Y_1 and Y_2 = components of the accelerating forces at the wrist- and crank-pins at right angles to X_1 and X_2;

P_a = force produced by the pressure of the steam on the piston;

F_1 = force required to produce the acceleration of the mass of the piston, piston-rod, and cross-head;

G and G_f = normal component of the reaction of the cross-head guides for frictionless and rough pins;

N_f and T_f = components of P_{cf} parallel and perpendicular to the crank;

T_f' = the latter reduced to the centre of the crank-pin.

We now have $C = W_1 \sin \delta.$

The parts of the weight of the connecting-rod supported at wrist- and crank-pins are

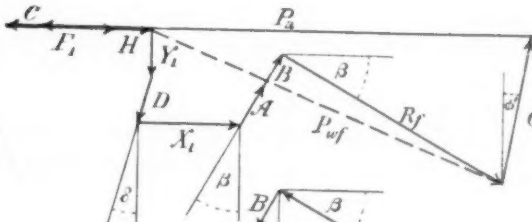
$$D = \frac{n - l + c \tan (\beta - \delta)}{n} W_2,$$

and

$$E = \frac{l - c \tan (\beta - \delta)}{n} W_2,$$

Combining these forces with P_a, the accelerating forces, and those due to friction, we obtain the equilibrium polygons, represented in Figs. 268 and 269, for the forces acting on the wrist- and crank-pins.

FIG. 268.



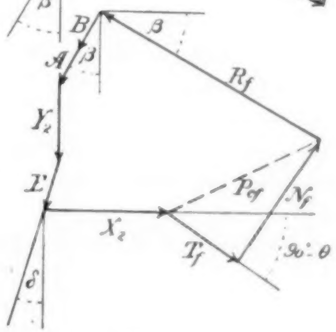


FIG. 269.

From these polygons we obtain the following values of the several forces:

$$G_f = \frac{1}{1 + \tan\varphi' \tan\beta}\left(\begin{pmatrix} P_a + C - F_1 - H + D\sin\delta \\ -(A+B)\sin\beta - X_1 \end{pmatrix}\tan\beta \atop + Y_1 + D\cos\delta - (A+B)\cos\beta\right)$$

$$P_{wf} = \sqrt{[(P_a + C - F_1 - H - G_f\tan\varphi')^2 + G_f^2]},$$

$$T_f = \begin{pmatrix} P_a + C - F_1 - H - G_f\tan\varphi' \\ + D\sin\delta - (A+B)\sin\beta - X_1 \end{pmatrix}\sec\beta\sin(\theta+\beta)$$

$$- Y_2\cos\theta - X_2\sin\theta - (A+B)\cos(\theta+\beta) - E\cos(\theta+\delta),$$

$$N_f = \begin{pmatrix} P_a + C - F_1 - H - G_f\tan\varphi' \\ + D\sin\delta - (A+B)\sin\beta - X_1 \end{pmatrix}\sec\beta\cos(\theta+\beta)$$

$$+ Y_2\sin\theta - X_2\cos\theta + (A+B)\sin(\theta+\beta) + E\sin(\theta+\delta),$$

$$P_{cf} = \sqrt{(T_f^2 + N_f^2)},$$

$$T'_f = T_f - Bn.$$

APPLICATION OF THE FOREGOING FORMULÆ.

We will apply these formulæ to the case of a horizontal high-speed engine, assuming an excessive coefficient of friction at the wrist- and crank-pins, and show that the values of A and B are determined by one approximation to as great a degree of accuracy as is desirable.

The dimensions of the engine are:

Length of stroke, 12″;

Diameter of cylinder, 10″;

Length of connecting-rod, 36″;

Distance from the wrist-pin to the centre of gravity of the rod, 20″.15;

Principal radius of gyration, 15″.00;

Diameter of crank-pin, 3″;

Diameter of wrist-pin, $2\frac{1}{2}$″;

Weight of piston, piston-rod, and cross-head, 90 lbs.;

Weight of connecting-rod, 70 lbs.;

Sin φ, .24.

Neglecting friction, the forces contained in Table I. are obtained.

TABLE I.

FORCES IN POUNDS PER SQUARE INCH OF PISTON AREA.

θ	P_a	F_1	Y_1	Y_2	X_1	X_2	G	T	N	P_w	P_c
0	+83	+20.50	0	0	+6.85	+7.82	0	0	+47.8	+62.5	+47.8
30	+81	+16.60	−.50	−3.33	+5.63	+6.71	+4.43	+33.1	+41.0	+64.4	+52.6
50	+76	+10.81	−.76	−5.10	+3.72	+4.80	+7.17	+51.7	+26.4	+65.6	+58.1
70	+54	+ 3.74	−.93	−6.26	+1.41	+2.49	+6.84	+48.4	+ 2.7	+50.7	+48.4
90	+33	− 2.97	−.99	−6.63	− .85	− .17	+5.20	+37.0	−12.9	+36.4	+39.2
110	+20	− 8.28	−.93	−6.26	−2.70	−2.74	+4.00	+27.9	−22.1	+28.6	+35.5
130	+ 1	−11.79	−.76	−5.10	−4.00	−4.95	+1.41	+12.0	−19.5	+12.9	+22.9
150	−25	−13.74	−.50	−3.33	−4.78	−6.54	−1.04	− 2.4	− 1.4	−11.3	− 2.7
180	−80	−14.64	0	0	−5.17	−7.48	0	0	+52.7	−65.4	−52.7

As a first approximation: *i.e.*, letting

$$A_1 = \frac{P_w r_w \sin\varphi}{nR} \text{ and } B_1 = \frac{P_c r_c \sin\varphi}{nR},$$

we obtain the values of A and B given in Table II.

TABLE II.

FORCES IN POUNDS PER SQUARE INCH OF PISTON AREA.

θ	A_1	B_1	$A_1 + B_1$	θ	A_1	B_1	$A_1 + B_1$
0	+ .521	+ .478	+ .999	110	− .238	+ .355	+ .117
30	+ .537	+ .526	+ 1.063	130	− .108	+ .229	+ .121
50	+ .547	+ .581	+ 1.128	150	− .094	+ .027	− .067
70	+ .422	+ .484	+ .906	180	− .545	+ .527	− .018
90	0	+ .392	+ .392				

Making use of the values of A and B given in Table II., we obtain the forces when friction is included as given in Table III.

TABLE III.*

FORCES IN POUNDS PER SQUARE INCH OF PISTON AREA.

θ	G_f	$G - G_f$	N_f	$N - N_f$	T_f'	$T - T_f'$	P_{cf}	$P_c - P_{cf}$	P_{wf}	$P_w - P_{wf}$
0	− 1.0	+ 1.00	+ 47.8	0	− 3.9	+ 3.88	+ 47.8	− .01	+ 62.5	+ .01
30	+ 3.4	+ 1.06	+ 41.5	− .53	+ 29.0	+ 4.08	+ 52.5	+ .13	+ 64.4	+ .06
50	+ 6.0	+ 1.13	+ 27.3	− .87	+ 47.6	+ 4.22	+ 57.9	+ .18	+ 65.5	+ .11
70	+ 5.9	+ .92	+ 3.6	− .86	+ 45.2	+ 3.21	+ 48.2	+ .24	+ 50.6	+ .11
90	+ 4.8	+ .40	− 12.5	− .40	+ 34.6	+ 2.55	+ 39.1	+ .13	+ 36.4	+ .05
110	+ 3.8	+ .12	− 22.0	− .11	+ 25.8	+ 2.06	+ 35.5	+ .07	+ 28.6	+ .05
130	+ 1.3	+ .11	− 19.4	− .09	+ 10.7	+ 1.28	+ 22.9	+ .04	+ 12.9	+ .01
150	− .9	− .09	− 1.4	+ .03	− 2.5	+ .21	− 2.7	.00	− 11.3	.00
180	.0	+ .02	+ 52.7	0	− 3.0	+ 3.18	− 52.7	.00	− 65.4	.00

Let the values of A and B as determined by a second approximation be A_2 and B_2; then will

$$A_1 - A_2 = \frac{(P_w - P_{wf})r_w \sin \varphi}{nR},$$

and

$$B_1 - B_2 = \frac{(P_c - P_{cf})r_c \sin \varphi}{nR}.$$

Table IV. contains values of $A_1 - A_2$ and $B_1 - B_2$.

* The computation of this table is facilitated by employing the following equations:

$$G - G_f = (A + B) \sec \beta,$$

$$N - N = (A + B) \tan \beta \cos (\theta + \beta) - (A + B) \sin (\theta + \beta),$$

$$T - T_f = (A + B) \tan \beta \sin (\theta + \beta) + (A + B) \cos (\theta + \beta) + Bn,$$

$$P_w - P_{wf} = \frac{P_w^2 - P_{wf}^2}{P_w + P_{wf}},$$

$$P_c - P_{cf} = \frac{P_c^2 - P_{cf}^2}{P_c + P_{cf}}.$$

TABLE IV.

FORCES IN POUNDS PER SQUARE INCH OF PISTON AREA.

θ	A_1-A_2	B_1-B_2	θ	A_1-A_2	B_1-B_2	θ	A_1-A_2	B_1-B_2
0	.0001	.0001	70	.0009	.0024	130	.0001	.0004
30	.0005	.0013	90	.0004	.0013	150	.0000	.0000
50	.0009	.0018	110	.0004	.0007	180	.0000	.0000

The greatest difference between the values of A and B, as obtained by a first and second approximation, is for $\theta = 70^\circ$, at which position $(A_1 + B_1)-(A_2 + B_2) =$ .0033 lbs. per square inch of piston area, or .0036 $(A + B)$.

The effect of such an error in $A+B$ upon the result may be found as follows:

The introduction of the forces $(A+B) = .905$ for $\theta = 70^\circ$ alters the value of P_c by the amount of .24 lbs. per square inch of piston area; a variation of $A+B$ equal to .0036 $(A+B)$ will, therefore, cause this difference in P_c to vary about .0036 times itself, or .0009 lbs. per square inch of piston area. This is equivalent to a variation in P_{cf} of .000019 P_{cf}, *i.e., the value of P_{cf} as obtained by one and two approximations differs by about* .00002 *times itself*. The variation in P_{wf} for $\theta = 70^\circ$, obtained by a similar method of reasoning, is about .000007 times itself. It is evident, therefore, that the value of $A + B$ as determined by the first approximation is sufficiently accurate.

CCCCXII.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR

HORATIO ALLEN.

[NOTE.—The unique relation borne to the American Society of Mechanical Engineers and to the profession of mechanical engineering in America by Mr. Horatio Allen, made some more extended notice seem fitting and desirable, than the brief memorials usually prepared for the Society's volume. In this view, the masterly and appreciative sketch prepared by Mr. M. N. Forney, member of the Society, for the *Railway and Engineering Journal*, is by his kind consent here reproduced, as a paper, No. 413, entire, except only the cuts and some details relating to the controversy as to the invention of the car truck principle, for which those interested are referred to the original document. Thanks are hereby returned to Mr. Forney for his courtesy.—*Secretary.*]

GUSTAV ADOLPH HIRN.

M. HIRN, honorary member of this Society, was born August 21, 1815, at Logelbach, near Colmar, in Alsace. His father was connected with an important cotton factory and allowed his son, when quite young, to work in the laboratory in which the colors were prepared for the dyeing and printing department. His health, when a boy, precluded his receiving much school education. When, some years later, this laboratory was closed, M. Hirn took charge of the mechanical department of the mill, a post which he retained until 1880.

The labors of M. Hirn in scientific life were manifold, both in physics and in metaphysics. Perhaps those for which he is most famous among engineers were his experiments on the steam-engine, about 1850, which were only terminated by the break-down of the historic engine in 1878.

His first publications were an "Essay on the Mathematical Theory of Fan-Blowers" (1845), "A Notice on the Gauging of Streams" (1846). Later, he began his experiments on friction, and completed them in 1847, but they were not published until 1854. These experiments were particularly on lubricated or "mediate" friction, and indicated that the coefficient varies

approximately as the square root of the pressure and as the exposed surface, and the first power of the speed; it decreases within limits, with the increased fluidity of the lubricant. The extent of this investigation has been lost to sight in later days from the fact that it was only recorded in periodical literature of the time, and has never been given a place in treatises on the subject, until now much more extended and complete investigations have thrown it into the background. Hirn also repeated the Meyer and Joule experiments, working independently of those physicists. It was by these investigations that his attention was especially directed to the subject of heat. A long list of scientific publications was the result of this determination of the direction of his scientific thought, many of which publications secured prizes to their author from various scientific bodies. In 1854 and 1855 Hirn brought out his papers on the "Interpretation of the Phenomena caused in a Steam-Engine by the Presence of the Jacket." He pointed out the fact, essential to Carnot's theory, and as developed by Clapeyron, as to the disappearance of heat in the engine, in consequence of its conversion into the dynamic form of energy. He also showed that the animal system, as a heat-engine, is a machine of high efficiency as compared with those constructed by man. His study of the steam-engine was conducted, for the first time, in so systematic and thorough a manner as to show precisely how energy is therein distributed and how heat is disposed of; and thus, for the first time also, revealed the method and extent of the wastes of the modern engine. He further showed how those wastes may be ameliorated by the use of either the steam-jacket or of superheated steam; showed that the former is superfluous if the latter is used; and exhibited the true action of the jacket, then entirely misunderstood by the majority both of men of science and of engineers.

It is a fact not without interest that the experimental engine at Logelbach was altered so as to have four valves, as long ago as 1856, before the Corliss construction had been introduced into Europe. The result of these investigations was to induce M. Hirn to employ superheated steam first on a considerable scale in 1856. His results were confirmed in a "Report" in 1869, by Dwelshauvers-Dery, Hallauer and Grosstete, before the Société Industrielle de Mulhouse.

In addition to the products of M. Hirn's industry in the field

of thermodynamics and steam-engineering, he also gave considerable attention to meteorology and astronomy. His mind was of an inventive turn as well as abstractly scientific, and he devised or modified a number of pieces of apparatus for experimental use ; among them was a form of typewriter, as early as 1855, and a power-measuring device, which he called a Pandynamometer, and others. His brother Ferdinand was the proposer of the scheme for transmitting power to a distance by rope.

A very full account of the scientific work of M. Hirn by M. Dwelshauvers-Dery, honorary member of this society, may be found in *London Engineering*, March 12, 1890. His principal papers may be read in the "Bulletin de la Société Industrielle de Mulhouse" and in his published works.

M. Hirn's mind remained active almost to the day of his death, his "Constitution de l'Espace Céleste" being published as recently as 1888. His health had been failing for some months previous to his decease, which took place January 14, 1890, at the age of seventy-four. He was made an honorary member of this Society at the New York [VIth] Meeting, November, 1882.

JOHN COFFIN.

Was born September 18, 1856, at Chatham, N. Y. His apprenticeship was served with Henry Clark of that place, on repair work, mostly on paper-mills. Thence he was called to the charge of a saw-mill in the Indian Territory, and his mechanical expedients were severely taxed where pipe-joints had to be made without dies, and where injector nozzles were made from cartridge-shells. He went one year to Cornell University. He worked as machinist in Ithaca and Syracuse, in the latter place becoming foreman for the Straight Line Engine Company, and acting as tool-maker for the Victor Sewing Machine Company. In 1881 he went to the Cambria Iron Company as draughtsman, foreman, and engineer. His work with this company was of particular interest and importance in the line of mechanical treatment of steels. He worked at this problem, and finally devised a method of treating forged axles which made certain the production of reliable work. He discovered in his investigations two singular properties of steel : the welding of surfaces at a temperature lower than carbon changes can occur, and the yielding of steel by its own weight, at that temperature. His latest important work

was upon the annealing and galvanizing of wire, for which he designed special machinery, which had just been put in operation last May, before the Johnstown disaster. Upon the death of Alexander Hamilton, Jr., he became chief draughtsman and designer at the Cambria Iron Works.

His death, September 3, 1889, from typhoid fever, was directly traceable to the effect of the calamity on his city. He joined the Society at the XVth or Washington Meeting of 1887, and had presented two papers to the Society, one handed in by Professor Sweet, descriptive of his averaging instrument, and the other giving the results of some of his investigations on steel.

HOWELL GREEN.

Mr. Green was born in Phillipsburg, N. J., November 17, 1830. At the age of seventeen he was apprenticed to the firm of W. & A. Dehaven, of Minersville, Pa., in whose shop he became foreman and afterward superintendent. For a number of years he was shop-foreman of the shops of the Dickson Manufacturing Company of Scranton, and since 1875 was superintendent of the Jeanesville Iron Works, with which he was connected at the time of his death. He became a member of the Society in 1881. His death, on October 25, 1889, was due to Bright's disease.

FREDERICK B. RICE.

Was born March 11, 1851, at Cincinnati, Ohio. At the age of sixteen he entered the machinist's trade, at the Shepard Iron Works, Buffalo, N. Y. In 1873 he was draughtsman and salesman for Brown & Struthers' Iron Works, and was for a time their master-mechanic. In 1878 he became superintendent of the Dunkirk Iron Works, and of the firm of Rice & Fell of that city. In 1885 he entered the employ of J. T. Noye, of Buffalo, as superintendent of the steam-engine department, where he remained until the failure of his health.

He invented an oil engine, which was manufactured and put on the market, and also designed the Rice Automatic Compound Engine, built by the firms with which he was connected. His death was due to consumption, and took place December 6, 1889, at Dunkirk, N. Y. He became a member of the Society at the Erie [XIXth] Meeting, May, 1889.

HECTOR C. HAVEMEYER.

Was born in New York City in 1840. He was educated at Union College, which he left in 1861, before graduating. Soon after this time he went to Germany and studied the principles of the sugar business, returning in 1865 to associate himself with the sugar house of Harris & Dayton. Afterward, with Mr. Albert Havemeyer, his uncle, a model house was erected in North Second Street, Williamsburg, which later became one of the houses controlled by Havemeyer & Elder.

He started, in 1871, another house in Jersey City, under the names of Havemeyer, Eastwick & Co., the two latter being afterward united under The Havemeyer Sugar Refining Company, of which Mr. Havemeyer was president.

He connected himself with the Society only at the Erie [XIXth] Meeting, and constituted himself at once a life member. He died in Paris, December 14, 1889, as the indirect result of overwork in connection with his business.

CHARLES A. ASHBURNER.

Mr. Ashburner graduated at the head of his class, in 1874, in the Towne Scientific School of the University of Pennsylvania.

His first work was as "transit-man," locating range-lights for the United States Light-House Board, along the Delaware River, below Philadelphia. From 1875 to 1885 he was engaged in geological work in the State of Pennsylvania, first in Huntingdon County and later in charge of four other counties. In 1880 he was in charge of the survey of the entire anthracite coal field, and in 1885 in charge of the survey of the entire State. During this time he was also frequently employed as expert in the location of mining fields, gave advice on the subject of mine-working and exploration for oil. In 1886 he became Engineer and Geologist of The Fuel, Gas and Electrical Engineering Company of Pittsburg, with which he remained connected up to the time of his death.

He contributed over twenty memoirs on engineering subjects to the technical societies, and was author, either in whole or in part, of eighteen of the "Reports" connected with the survey of Pennsylvania. He died December 24, 1889, having connected himself with the Society at the XVth or Washington Meeting of 1887.

CHARLES D. SMITH.

Was born in Plantsville, Conn., February 19, 1855. He graduated in the Class of '77 at the Sheffield Scientific School of Yale College, in the course of Dynamical Engineering.

His business and professional connection was principally with the firm of H. D. Smith & Co., Manufacturers of Carriage Hardware, his work being the designing and arrangement of machinery and shops, and general supervision over the company's affairs. At the time of his death he was also manager of The Connecticut Electric Motor Company, treasurer of the Southington Board of Trade, and a director of its National Bank. His death occurred very suddenly, January 23, 1890, from the effects of a fall. His life was very interesting as an example of the conquest of will and determination over the limitations imposed by physical disability.

EDWARD H. OWEN, JR.

Was a graduate in 1879 of the Massachusetts Institute of Technology. After graduation he had some practice in experimental testing of stationary steam and pumping engines, and in setting up such engines, and was also employed in testing at Holyoke on turbines. He entered the Society as a junior in 1880 when he was acting as assistant to E. W. Thomas, with the Willimantic Linen Company, upon their buildings and other work. He was for some five or six years with the Lowell Machine Shop, and in 1886 was appointed superintendent of the Atlantic Mills in Lawrence. He was able to retain this post but a few months, when on account of his health he was compelled to go to Colorado. He died at Colorado Springs, July 3, 1890.

HENRY J. DAVISON.

Mr. Davison was born in New York City, December 22, 1835. After leaving school he entered an insurance office, but remained there only a year, to become an apprentice at the Chelsea Iron Works, at the foot of West Twenty-sixth Street, then owned by Messrs. Meade and Ayers, and building gas-works and light-draught steamers. He remained here for seven years, three years until the old works were sold out (to the Novelty Iron Works mostly), and the rest of the time at the reorganized works in the shops at Fifty-ninth Street. He passed through the machine and

pattern shop, the drawing-room and ship-building departments, in course, and was for some time superintendent.

Mr. Davison took the contract for the taking down of the dome of the New York Crystal Palace, after its destruction by fire, and carried the work through successfully. He became thus specially conversant with iron structural work, and was for many years connected with contractors' work in South America, making a specialty of light steamers and iron structures which were built here and shipped for assembly on the ground. He was contractor for the telegraph installations for the United States of Colombia.

For the last fifteen years he made a specialty of gas-engineering and contracts. He took much interest in the water-gas methods of manufacture, and was concerned in the erection of some of the largest works in New York, Brooklyn, Baltimore, Chicago, Albany, and Syracuse, and was part-owner in many successful companies.

He died of heart failure, on the steamer which was bringing him home from Europe, July 22, 1890. He became a member of the Society at the Cleveland Meeting in June, 1883.

CCCCXIII.

A MEMORIAL OF HORATIO ALLEN.

BY M. N. FORNEY, NEW YORK CITY,

(Member of the Society.)

At the close of the last day of the year 1889, Horatio Allen, one of the oldest engineers in the world, passed away from earth. His life began with the opening of the century, and was identified with many of its earlier engineering enterprises, and he took a leading part in the introduction of railroads and locomotives in this country.

He was born in Schenectady, N. Y., in 1802, and was consequently eighty-seven years old last year. He was the son of Dr. Benjamin Allen and Mary Benedict Allen. His father was professor of mathematics at Union College, at Schenectady, and afterward established a large school at Hyde Park, N. Y. The son therefore had excellent educational advantages in early life, and was sent to Columbia College in New York City, from which he graduated about 1823, taking high rank in mathematics. He studied law for about a year, but after a short time decided to make engineering his work, and entered the employment of the Chesapeake & Delaware Canal Company under Judge Wright, then Constructing Engineer of that work. He was sent to St. George's, Del., and within two weeks was placed in full charge of a party. In the autumn of 1824 he was made Resident Engineer of that work. A year later he was appointed Resident Engineer of the summit level of the Delaware & Hudson Canal, under John B. Jervis, then Chief Engineer of the company. While Mr. Allen was engaged in this position his attention, and that of other engineers in America, was attracted to the performance of locomotives in England. His early relation to these events may best be told in his own words, quoted from a pamphlet with the title "The Railroad Era," which he published a few years ago. In this he said:

"During the years 1826 and 1827 the use of the locomotive on the Stockton & Darlington Road, England, had become known to many, and especially to civil engineers in this country,

and among others to myself, then a Resident Engineer on the line of the Delaware & Hudson Canal, the great engineering enterprise of the time, the first of the great works, canal and railroad, that were to bring the anthracite coal of the valley of the Susquehanna into the valleys of the Delaware and of the Hudson and to the ocean.

"Such consideration as was within my power led me to a decided conviction as to the future of the locomotive as the tractive motive power on railroads for general freight and passenger transportation, as it had begun to be for mine transportation. . . .

"Early in the year 1827 I had given all the attention that it was in my power to give, and having come to conclusions as to the locomotive, that all subsequent experience has confirmed, and believing that the future of the civil engineer lay in a great and most attractive degree in the direction of the coming railroad era, I decided to go to the only place where a locomotive was in daily operation and could be studied in all its practical details.

"Closing my service on the Delaware & Hudson Canal, some two months were appropriated to certain objects and interests, after which I was again in New York, preparatory to going to England.

"On my return to New York from these visits I found that it had been decided by the Delaware & Hudson Canal Company to intrust to me, first, the having made in England for that company the railroad iron required for their railroad . . . and having built in England, for the company, three locomotives, on plans to be decided on by me when in England.

"This action of the Delaware & Hudson Canal Company was on the report of their Chief Engineer, John B. Jervis, and thus it occurred that the first order for a locomotive engine, after the locomotives on the Stockton & Darlington Road were at work, came from an American company, on the report of an American civil engineer."

The following are copies of some old papers, which were preserved by Mr. Allen, relating to this commission. One of them is indorsed :

"1828.—EXTRACTS FROM THE REPORT OF THE COMMITTEE SANCTIONED BY THE BOARD REFERRED TO IN MY LETTER TO MR. ALLEN.

'J. BOLTON, *President*."

The following is a copy of the paper, which evidently embodies the instructions of a committee to Mr. Allen for the execution of his commission in England:

EXTRACT, ETC.

That Horatio Allen, Esq., Civil Engineer, has agreed to go to England as the agent of the company, to procure the railroad plates and perform such other services in relation thereto as may be required of him. The company to pay his passage out and home and his expenses during his stay, allowing him to remain three months, for the purpose of attending to the company's business and acquiring information. His expenses on the whole not to exceed $900, and on his return he will communicate to the company all the information he may acquire that may be useful to the work in which they are engaged.

That they deem it advisable to authorize Mr. Allen to procure one locomotive engine complete, as a pattern, and that the Chief Engineer is making inquiries to ascertain whether it may not be expedient to authorize the construction of all the locomotive engines in England.

That it is deemed advisable to suspend the making of the wheels and axles of the coal wagons until information be received from Mr. Allen of the cost of those articles in England, and of the latest improvements that have been adopted in the manner of connecting the wheels and axles. The engineer in his report recommends wheels of 3½ feet diameter; but his mind is not definitely made up on this point. He will investigate the matter further and report the result.

That Mr. Allen be instructed to procure the railroad plates of the length recommended in the report of the Chief Engineer; the ends to be cut and fitted into each other and the holes made for the fastenings, as recommended in the same report; that the rounding of the edges of the plates will be advantageous, but is not so indispensable as to induce the committee to recommend that the plates be thus formed without limitation as to the increase of expense and time that may be required therefor.

That there is much force in the reasoning of the Chief Engineer in favor of dispensing with any allowance for expansion and contraction of the plates, in forming the holes for the fastenings, yet the committee are of opinion that it would be safest to make such allowance, and the Chief Engineer has devised a plan for effecting it which the committee believe will be successful. This plan will be communicated to Mr. Allen, and he may then be allowed very safely to adopt that or any other plan which may be found more economical.

The committee being now satisfied that an economical plan will be devised for forming the holes in the plates so as to allow for contraction and expansion, they unite in opinion with the Chief Engineer that the fastenings of the plates will be best effected with screws.

That the Chief Engineer is of opinion that on two of the levels west of the summit and one east of the summit, machinery worked by engines may be advantageously substituted for the horse-power first proposed, but that the form of the country will not admit of such substitution on the other levels west of the summit.

S. Flewelling,

Treasurer of the Delaware & Hudson Canal Company.

EXTRACT FROM THE MINUTES OF JANUARY 10, 1828.

The rounding of the edges or omitting it is left to Mr. Allen. The last paragraph is introduced to suggest to Mr. Allen that information is wanted. In the letter to Messrs. Brown I say : "Mr. Allen is authorized to procure such drawings of machinery and designs connected with railroads, canals, and the raising and transporting of coal as he may deem proper."

J. BOLTON.

A letter from Mr. Jervis, the Chief Engineer, which apparently accompanied the preceding "Extracts from the Report of the Committee," is indorsed :

"1828.—MR. JERVIS'S LETTER TO MR. ALLEN REFERRED TO IN MY LETTER TO MR. ALLEN.

"J. BOLTON."

The letter is as follows :

To Horatio Allen, Esq.:

DEAR SIR : The Board of Managers for the Delaware & Hudson Canal Company, having made an engagement with you to proceed to England as their agent to procure certain articles for the proposed Carbondale Railroad, and also such information as may be useful in the construction and management of said railroad ; I am therefore directed by the said Board of Managers to furnish you such information and instruction as will further their object.

The Board of Managers have determined on procuring their iron plates for the railway tracks as one item.

The length of plates to be from 12 to 14 feet, as you may find most convenient for rolling them through on the edges; to be $2\frac{1}{4}$ inches wide on the bottom and 2 inches on the upper surface and $\frac{1}{2}$ inch thick, with the upper edges rounded and the end finished as represented on the plan. Holes to be drilled for the screws with countersunk heads at each end of every bar and at intermediate points 18 inches apart. After the holes and countersink have been drilled in a circular form, then a rimmer of the proper form to fit the countersink and hole for the neck of the screw to be put in to cut the aperture longitudinally. To effect this the rimmer must be put in and then firmly fixed to its position and the bar made to move toward it in the direction of its length, about $\frac{1}{8}$ or $\frac{1}{4}$ of an inch. This may be reduced as you recede from its end to the centre ; but as it is likely to create confusion to attempt any economy in varying the length to be rimmed, it will be better to have all the holes rimmed alike.

LOCOMOTIVE STEAM ENGINES.

It is desirable, in order to dispense with the tender carriages, to have a water-tank fixed to the engine-carriage that will contain about 100 gallons. If made in two parts, of sheet iron, it will weigh, with its hanging or supporting irons, about 250 lbs., and the water about 1,000 lbs., making together about 1,250 lbs. To increase the capacity of the tank to 120 or 150 gallons would add but little to its weight. I see no difficulty in attaching such a tank to the engine-carriage, and you will determine whether it will be most convenient to support it over the axles or suspend it under them ; being divided into two equal parts it may be placed on each side of the revolving chains, with a pipe to pass the water from

one to the other. If the weight of the engine should admit of it, it will be preferable to make the tank sufficient to contain 120 to 150 gallons. The pump of the engine to supply the boiler with water from the tank should be calculated to work one-quarter faster than necessary for a regular supply in order to provide for a waste of steam when the engine stops, and to be constructed so as to work by hand, which will be necessary at certain times. The boiler will not require a capacity for any considerable quantity of water beyond what is necessary for the work, as the pump will regularly supply, except when the carriage stops at the end of the road, at which time it is supposed to have supplied a surplus adequate to the waste that will take place during delay. The stoppages may be estimated at one-quarter the whole time ; on the shortest section 10 minutes, on the longest 20 minutes. The weight of engine, carriage, and water, if placed on six wheels, to be from 6 to 7 tons, but 6½ tons preferred. If it should be found that a six-wheel carriage has any important difficulty in working well on curved roads, that in your judgment would counterbalance the advantage of a heavier engine and give the preference to the four-wheel carriage, then the weight must not exceed 5½ tons ; but the six-wheel carriage will be preferred if it can be made to work. If a six-wheel carriage the axle need not exceed 2¾ or 3 inches at the bearing. The diameter of the wheels 3 to 4 feet, as you find most approved from experiments in England for similar purposes and rate of travelling, say 3½ to 5 miles per hour. The power of the engine such as will carry 800 lbs., at the rate of 4 miles the hour, or what is nearly the same thing, 640 lbs., at the rate of 5 miles the hour. I think about 4 miles the hour a good velocity for the work contemplated, but the range above given will allow you to vary this as you may find most expedient, in relation to several points that you will perceive to have a bearing on this question. The diameter of the wheels of the engine-carriage will affect the velocity, or distance travelled at a given number of strokes of the engine, but I would take 3 feet as the minimum diameter and make them as much larger as the arrangement of the working parts will admit, without giving too great a velocity. The length of the stroke must depend something on the facilities of securing firmness to the cylinder, and this may lead you to prefer a larger or smaller diameter for the cylinder ; the pressure of the steam has also a bearing on the question; on account of the weight I think the cylinder should not exceed 8 inches.

To elucidate my views more fully, I will state what appears to me a suitable arrangement. Length of stroke 27 inches and 40 strokes per minute; two 8-inch cylinders, pressure of steam 60 lbs. per square inch. This will give 2,400 revolutions per hour. Area of cylinder $8^2 \times .7854 \times 2 = 100.5$ square inches. A double stroke equal 4.5 feet; then $100.5 \times 4.5 \times 60 \times 2{,}400 = 65{,}124{,}000$ lbs. raised 1 foot. But by the experiments of Wood we may only take 30 per cent. and $65{,}124{,}000 \times .30 = 19{,}537{,}200$ raised 1 foot, which is equivalent to 800 lbs. carried or raised 24,421 feet, equal to 4.62 miles per hour. Now $24{,}421 \div 2{,}400 = 10.17$ feet, the space moved over by the carriage at each revolution of the engine, and of course the diameter of the wheel must be 3.25 feet. If there should appear a difficulty in securing a cylinder with proper stability for the above length of 27 inches, it may be advisable to make the stroke 25 inches. Then, all other points remaining the same, the power of the engine will only be equal to carry the same load of 800 lbs. 4¼ miles, and the wheels of the carriage must be reduced to 3 feet diameter. It may be found expedient to have larger wheels and travel at the rate of 5 miles per hour with a proportional load. Suppose cylinder 7½ inches diameter, 27 inches stroke, pressure 60 lbs., 40 double strokes per minute,

or 2,400 per hour. Then $7.5^2 \times .7854 \times 2 = 88.34$, say area of two cylinders $88 \times 60 \times 4.5 \times 2{,}400 = 57{,}224{,}000$ lbs. raised 1 foot; 30 per cent. is 17,107,200 lbs. raised 1 foot, and is equivalent to 648 lbs. carried 5 miles, will require the carriage wheel to be 3½ feet diameter. The power of the engine will be a trifle less than the last calculation before it. But if you find it necessary to reduce the length of stroke to 25 inches it will not give the power we prefer with less than 8-inch cylinders. If, as before observed, you find difficulties that have not been anticipated in working a six-wheel carriage, that compels the use of a four-wheel carriage, the power of the engine must be reduced in order to reduce the weight of the engine and the carriage. If you can avoid it, I think it better not to calculate for more than 40 double strokes per minute. I believe the above will give you a sufficient view of what will answer our object, and you must vary as you find the experience of England and your own judgment may direct. I am of opinion that the furnace had better be of the oval form laid flat, otherwise the furnace may be the same as for bituminous coal. It is supposed anthracite coal does not require so high a chimney as other fuel, but I am not possessed of any particular facts on this subject; I presume you can have the chimney so constructed that an additional piece may be attached if it is found on trial to require it. On this presumption I would not have it more than 10 feet high. As the height of chimney will affect the calculation of bridges, it is advisable to understand this question as early as possible.

The width in the clear between the rails is 4 feet 3 inches. The greatest curvature of that part of the road on which locomotive engines are to be used is that which gives a versed sine of 1 foot on a chord of 59 feet; but there is only a single instance of this curvature, arc of 15 chains. The curvature which occurs in several instances is a versed sine of 1 foot on a chord of 66 feet. A 10-foot chord gives exactly ⅓ inch.

It is determined by the Board that you will procure from England one locomotive engine with carriage complete for work. The three others that will be wanted to depend on the cost at which they can be obtained and delivered at New York. It is supposed that they can be obtained of American manufacturers for $1,800, and I presume it will not be economy to procure them from England at a greater cost, unless you perceive a superiority in the workmanship of English engines that in your opinion will justify the additional cost.

As a preliminary step I should advise, previous to the purchase of the locomotive steam engine, that you visit the Killingworth Railroad, near Coventry, the Hetton Railroad, and Darlington & Stockton Road; the two latter are near Sunderland. At Killingworth the locomotive engine is said to have been in regular use (working by the adhesion of the wheels) since the year 1814; but the Hetton Road is more in the character of the proposed work.

Although I am strongly of opinion that this will be the most convenient and economical power for the contemplated railroad, still you will perceive the propriety of availing ourselves of the experience of others in reference to its actual utility. If on examination you should find essential difficulties that we have not apprehended in the use of this means of transportation, and such as in your judgment would counterbalance their advantages, then it will be advisable not to make an engagement, but to communicate the result of your observations as early as possible.

RAILROAD CARRIAGES.

Inquire respecting the relative advantages of the fixed and revolving axle of common railway carriages; their operation on curved roads; the methods and

facility of applying the brake; the manner of constructing and securing the axle to the wheel in both cases; facilities for oiling; the width of rim or track of wheels as compared to the width of rail; thickness, depth of projection and form of flange; breadth and thickness of spokes of cast and wrought iron; manner of handling and fastening the door in the bottom of the carriage to facilitate unloading coal.

It is deemed advisable to ascertain the cost of iron axle-trees for the coal-carriages, made of iron equal in quality to Swedes or Russia iron, the bar 2¼ inches square and the bearing 2¼ inches diameter turned smooth. State the cost distinct for revolving and fixed axles; as you will perceive, the fixed axle will, on account of its longer bearing in the nave of the wheels, require more expense in turning. Examine whether fixed axles are tapered from the shoulder to the outer bearing in the nave, or whether the axle is of uniform diameter through the nave of the wheel, and in what manner the wheel is secured to the axle, and box of carriage through the axle.

Very respectfully, your friend and obedient servant,

JOHN B. JERVIS.

NEW YORK, January 16, 1828.

The duties, say 27½ per cent. exchange, interest, and other charges, will together amount to about 45 per cent. on the cost of the engine.

"It was under these favorable circumstances," Mr. Allen says, "that I left New York in January, 1828, and within two days after my arrival at Liverpool I made the acquaintance of George Stephenson, in the most agreeable relations, and from that time, during my stay in England, I received from him every kindness in his power, and all the aid to what I had come so far to seek that was at his command at Liverpool, on the Stockton & Darlington Railroad, and at Newcastle, at that time the centre of all that was in progress in railroad and locomotive matters."

To get an idea of "the state of the art" of locomotive construction at the time Mr. Allen arrived in England, in 1828, it must be remembered that it was before the celebrated trial of the *Rocket* on the Liverpool & Manchester Railway, which did not occur until October 14, 1829. The form of locomotive engine which is described in Wood's "Treatise on Railroads," and which that author says, "with trifling modifications," was used on the Stockton & Darlington, the Killingworth, and other railroads in England, had cylindrical boilers, with hemispherical ends and a single cylindrical tube of about 2 feet diameter, which passed through the boiler and was placed within 2 inches of the bottom. In one end of this tube the fire was placed and the other end was terminated by a chimney. In some engines this tube, instead of passing through the boiler, was made to return and pass out at the same end as the fire-grate. The engines had

four wheels and two cylinders, which were placed vertically and attached to the top and partly within the boiler, and were located on the longitudinal centre line of the engine, one of them directly over each axle, the piston-rods working through the top cylinder heads with a long cross-head, which extended transversely far enough so that the connecting-rods could be coupled directly to crank-pins to the outside of the wheels. The cranks on the two pairs of wheels were at first maintained at right angles to each other by an endless chain passing over cog-wheels fixed upon the axles of the engine. That Mr. Jervis contemplated some such arrangement as this is indicated by the fact that he speaks of " revolving chains " in his letter. Of these chains Mr. Wood says :

"However good in other respects, this chain had its defects, and it has been superseded by cranks and connecting-rod. By continued working the chain was apt to stretch, and a contrivance was resorted to, of the removal of the chairs (?) from each other, to tighten the chain; but as this could only be done at certain periods, the chain was frequently getting slack. When this took place, and when the full power of one of the cylinders was applied upon one pair of wheels, while the other connecting-rod was upon the centre, and therefore not capable of acting at all upon the other wheels, the rotation of the latter depended upon the action of the chains ; if the chain was, therefore, slack, it occasioned a slipping of the wheel until the links of the chain laid hold on the projection of this wheel in the direction in which the chain was moving round, and this slipping alternately occurred by each of the wheels in succession, as they became a predominant moving power. The chain was therefore laid aside."

To maintain the cranks of the two pairs of wheels at right angles to each other, "returned cranks" were attached to the outer ends of the crank-pins of one pair of wheels. These returned cranks had what may be called secondary pins on their outer ends, which were placed at right angles to the main pin. These secondary pins were connected by coupling-rods to the main pins on the other pair of wheels, and thus the two sets main pins on the two pairs of wheels were kept at right angles to each other.

It should be observed that when Mr. Allen arrived in England the use of the multitubular boiler in locomotive engines was un-

known, or was only talked about. In the engraving of the Killingworth engine in Wood's "Treatise," he shows and describes an exhaust-pipe which "is opened into the chimney, and turns up within it;" but the value of the steam blast was then not recognized. The locomotives which were known in America at that date were those which have been described. It is therefore not remarkable that Mr. Allen, then only twenty-seven years of age, and feeling the responsibility of his position, should be governed by the instructions which he received when he left home. He therefore ordered of Messrs. Foster, Rastrick & Co., of Stourbridge, three locomotives of the Stockton & Darlington type.* One of these was the engine that afterward had the distinction of being the first one that was ever run in America. It had four coupled wheels, all drivers, driven by two vertical cylinders, with 36-inch stroke, placed at the back end and on each side of the boiler. The motion of the piston was transferred through two grasshopper beams above the cylinders, and from those beams by connecting-rods to the crank-pins on the wheels. The front end of the beam was supported by a pair of radius rods which formed a parallel motion. The spokes of the wheels were heavy oak timbers, strengthened by an iron ring bolted to the spokes midway between the hub and felloes, and the latter was made of strong timber capped by a wrought-iron tire. From the illustrations of this engine which have survived, the cranks on each pair of wheels were apparently at right angles to each other, otherwise it is not clear how the engine could start when they were on one of the dead points. The boiler was cylindrical and had several large flues inside.

After Mr. Allen arrived in England, as already stated, he made the acquaintance of George Stephenson, and from him received much valuable aid and advice. He visited Liverpool, the Stockton & Darlington Railway, and Newcastle. Locomotive

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* In the latter part of his life, Mr. Allen was of the impression that one locomotive was ordered of this firm and two of Messrs. Stephenson & Co., of Newcastle, but an examination, since his death, of some correspondence on file in the office of the Delaware & Hudson Canal Company has shown conclusively that three engines were built by the first-mentioned firm and one by the Messrs. Stephenson. This correspondence shows that the locomotive built by the Stephensons arrived in New York on board the ship *Columbia* about the middle of January, 1829. The first one of those built by Foster, Rastrick & Co. arrived on board the *John Jay*, May 13 of the same year; the second one on the ship *Splendid*, about the middle of August, and the last one on September 17, on the *John Jay*.

engines had then been in successful use since 1814, and the subject of railroads was attracting great attention, not only in that country, but in America and the whole civilized world as well.

The following extract from a report of the Second General Meeting of the Liverpool & Manchester Railway, dated March 27, 1829, will show how the subject was then regarded. In this report it is said:

The nature of the *power* to be used for the conveyance of goods and passengers becomes now a question of great moment, on whatever principle the carrying department may be conducted. After due consideration the engineer has been authorized to prepare a locomotive engine, which from the nature of its construction, and from the experiments already made, he is of opinion will be effective for the purposes of the company, without proving an annoyance to the public. In the course of the ensuing summer it is intended to make trials on a large scale, so as to ascertain the sufficiency, in all respects, of this important machine. On this subject, as on every other connected with the execution of the important task committed to his charge, the directors have every confidence in Mr. Stephenson, their principal engineer, whose ability and unwearied activity they are glad of this opportunity to acknowledge.

On his arrival in England Mr. Allen found, as Mr. Wood in the preface to his "Treatise" says, that "The eyes of the whole scientific world were upon the great work of the Liverpool & Manchester Railway;" and as another writer of that period reported, "discoveries were daily made of new principles applicable to locomotives, and, extraordinary as they now are in their power and velocity, great improvements may yet be reasonably anticipated." In England Mr. Allen spent considerable time in visiting the different roads then in operation, and in studying the performance of the locomotives in use. The kind of power to be used on the Liverpool & Manchester Railway was regarded as a question of great moment. In the spring of the year 1829 the directors of that company sent a deputation of their body to visit the lines where different varieties of motive power were employed. The only conclusion which they came to appeared to have been, that, from the great amount of traffic anticipated upon the line, horses were inapplicable. The contest then being between locomotive and fixed engines, in order to determine which of the two was the most suitable for the purpose, the directors resolved to employ two practical engineers, Mr. James Walker and Mr. John W. Rastrick, to report which, under all circumstances, was the best description of moving power to be used. They reported against locomotive and in favor of stationary engines. Notwithstanding this report, the directors did not

feel themselves able to come to a decision on the subject—a leaning in favor of locomotive engines existing, it was said, in a majority of the directors.

Mr. Allen made a contract with Messrs. Stephenson & Co., of Newcastle, for one more locomotive. This engine, he said, was ordered to be identical in boiler, engine plan, and appurtenances to the celebrated *Rocket*.

When completed the four engines were shipped to New York and arrived there during the year 1829. The *Stourbridge Lion*, it is said, was sent from the foot of Beach Street, in New York, to Rondout, and thence reshipped by canal to the track at Honesdale, where it made its celebrated first trip. Some of the other engines were for a time stored in the warehouse of Messrs. Abeel & Dunscom on the east side of New York. One of them was there raised up so that its wheels were not in contact with the ground and was exhibited in motion with steam on as a curiosity to the public. The singular part of this is that it is not now known what ever became of these engines. All trace of them has been lost as completely as though they had been cast into the sea.

Why the *Stourbridge Lion* was sent to Honesdale and not the Stephenson engine, which arrived in New York first, is not known. If this one, which has since passed into oblivion, had been selected for the first run, we would have had the remarkable circumstance that a trial of an engine which Mr. Allen said was built on substantially the plan of the famous *Rocket*, would have occurred in this country *before that* celebrated event took place in England.

"It is to be regretted," said Mr. Allen, "that one of the Stephenson locomotives was not sent, and for the reason that they were the *prototypes* of the locomotive *Rocket*, whose performance in October of the same year so astonished the world. If one of the two engines in hand ready to be sent had been the one used on August 9, the performance of the *Rocket* in England would have been anticipated in this country."

The story of this first trial of the *Stourbridge Lion* has often been told. The engine received its name, Mr. Allen said, "from the fancy of the painter, who, finding on the boiler end a circular surface, slightly convex, of nearly four feet diameter, painted on it the head of a lion in bright colors, filling the entire area."

The river and canal being closed by ice, it was not until the

opening of navigation in 1829 that access was had to the railroad at Honesdale, Pa., which was then at the head of the canal and at the beginning of the railroad.

Being at liberty during July and August, Mr. Allen volunteered to go to Honesdale and take charge of the transfer of the locomotive from the canal-boat to the railroad track. Of the place where the trial was made he wrote:

"The line of road was straight for about 600 feet, being parallel with the canal, then crossing the Lackawaxen Creek by a curve nearly a quarter of a circle long, of a radius of 750 feet, on trestlework about 30 feet above the creek, and from the curve extending in a line nearly straight into the woods of Pennsylvania.

"The road was formed of rails of hemlock timber in section 6 x 12 inches, supported by caps of timber 10 feet from centre to centre. On the surface of the rail of wood was spiked the railroad iron—a bar of rolled iron 2¼ inches wide and ½ inch thick. The road having been built of timber in long lengths, and not well seasoned, some of the rails were not exactly in their true position. Under these circumstances the feeling of the lookers-on became general that either the road would break down under the weight of the locomotive, or, if the curve was reached, that the locomotive would not keep the track, and would dash into the creek with a fall of some thirty feet.

"When the steam was of right pressure, and all was ready, I took my position on the platform of the locomotive alone, and with my hand on the throttle-valve handle, said: 'If there is any danger in this ride, it is not necessary that the life and limbs of more than one should be subjected to danger,' and felt that the time would come when I should look back with great interest to the ride then before me.

"The locomotive having no train behind it answered at once to the movement of the valve; soon the straight line was run over, the curve was reached and passed before there was time to think as to its being passed safely, and soon I was out of sight in the three miles' ride alone in the woods of Pennsylvania.

"I had never run a locomotive nor any other engine before, I have never run one since; but on August 9, 1829, I ran that locomotive three miles and back to the place of starting, and being without experience and without a brakeman, I stopped the locomotive on its return at the place of starting. After

losing the cheers of the lookers-on, the only sound, in addition to that of the exhaust steam, was that of the creaking of the timber structure.

"Over half a century passed before I again revisited the track of this first ride on this continent. Then I took care to walk over it in the very early morning, that nothing should interfere with the thoughts and the feelings that, left to themselves, would rise to the surface and bring before me the recollections of the incidents and anticipations of the past, the realization of the present, and again the anticipations of the future.

"It was a morning of wonderful beauty, and that walk alone will, in time to come, hold its place beside the memory of that ride alone over the same line more than fifty years before."

Mr. Allen always took a delight in telling of this early event in railroad history. When the enormous extent of the railroad system of this country is considered, it seems very wonderful that it was created within the lifetime of a single individual, who was an active, and, it may be said, the chief participant in the very beginning of steam locomotion in this country. Less than a year ago the venerable Captain John Ericsson ended his eventful life. He was a participant in the celebrated Rainhill trial of locomotives on the Liverpool & Manchester Railway in 1829. His life and that of Mr. Allen formed links which almost united the eighteenth and the twentieth centuries.

In September of 1829 Mr. Allen became the Chief Engineer of the South Carolina Railroad, the construction of which had then been determined upon. On his recommendation the gauge of the road was made 5 feet. This road was completed, and its cost was within his original estimates, and when finished it was the longest railroad in the world. Later, the question of the gauge of the Erie Road was referred to him as Consulting Engineer of that line. He advised that it also be made 5 feet. It was a great misfortune that his advice was not followed in both cases, and that the gauge for all American railroads was not made 5 feet. The extra width of 3½ inches would now be an immense advantage in the construction of both cars and locomotives. As the weight, size, and capacity of these have grown, the value of this 3½ inches of space between the rails has increased in about the same ratio.

At that early date the South Carolina Railroad Company had to decide whether the motive power of the road should be

horses or locomotives. In a report made to the company in November, 1829, Mr. Allen presented an estimate of the cost of transportation by horse-power and by locomotive power. The estimate of cost of locomotive power was based on facts obtained on the Stockton & Darlington Railroad. In his pamphlet, "The Railroad Era," Mr. Allen said:

> The result of that comparison was in favor of locomotive power, and the report contained a decided recommendation that locomotive power should be the power to be used on the South Carolina Railroad. But the basis of that official act was not the simple estimate resting on the facts as they existed on the Stockton & Darlington Railroad, but, as was stated in the report, was on the broad ground that in the future there was no reason to expect any material improvement in the breed of horses, while in my judgment the man was not living who knew what the breed of locomotives was to place at command.

This report was submitted to a full meeting of the board, and the decision was unanimous to adopt locomotives as the tractive power on the road, and Mr. Allen added, "It was the first action of this kind by any corporate body in the world."

The South Carolina Railroad when first constructed consisted of timber rails or stringers 6 x 12 inches, on which iron bars $2\frac{1}{2}$ x $\frac{1}{2}$ inches were spiked. When the question of motive power came up for consideration it was essential that the weight per wheel should not be greater than the structure described could safely bear. The load per wheel, therefore, had to be limited, and it also seemed to be highly important to place as great a quantity of power within one machine as possible. In another communication, made May 16, 1831, to the president and directors of the road, Mr. Allen discussed the general subject of steam transportation, and especially the subject of boiler capacity of locomotives, and then said:

> When we come to consider the application of locomotives to wooden roads there are circumstances which call for attention, and a particular adaptation of arrangement to them. As the same amount of attendance and repairs attend engines of the various powers within the range that can be employed on railroads, it becomes a highly important object to place as great a quantity of power within one machine as possible. And this is more peculiarly the case on a road where the great and most difficult sources of expense are the attendance and repairs, while the fuel is comparatively of little consequence. As on every road there exists a limit of weight to be placed on each pair of wheels, and as on wooden roads this limit is much less than on an iron one, it becomes a still more interesting inquiry to ascertain by what means we may increase the quantity of power without exceeding the limit. On the Liverpool & Manchester road they appear practically to be limited to three tons on each pair of wheels, though some accounts state this to be too high, with their velocity, for the permanent

benefit of the road. On a wooden road, where only ½-inch iron is made use of, I would put the limit at 1½ tons per pair of wheels.

If, therefore, there can be no arrangements whereby this disadvantageous relation may be provided for, it is evident that to convey the same quantity of goods or transport the same number of passengers, we must incur twice the expense of attendance, twice the amount of repair, and twice the liability to accident. In fact, more than twice, since in doubling the weight of the engine we are able to appropriate a greater proportion of the increased weight to steam-generating purposes.

The arrangement which I would propose to effect so desirable an object would be, as the limit exists in the quantity on each point of support, to increase the number of supports, and thus distribute the weight over a greater surface. I would therefore place the engine on six or eight wheels, and limit the weight to 1½ tons to each pair.

There arise two objections to this arrangement, from the inequalities in the line of support; the one vertical, the other horizontal.

If three or four wheels were united on a side to the same rigid straight line, and the road had irregularities in its surface, there would arise great and injurious strains to the structure, from the wheels not being able to adapt themselves to the irregularities.

This difficulty may be completely obviated by giving the weight to be supported but two points of support on each side, and making these points the centres of motion of the pairs of wheels.

This arrangement will evidently adapt itself with as much ease and simplicity to all vertical irregularities as is the case with two wagons connected together.

As to the change of direction horizontally, as in the entrance of turnouts and the passage of curves, a very simple adjustment will relieve the arrangement from all difficulty. If we connect the frame with the cross-piece only at the centre, and by a horizontal point, the two sets of wheels will thereby be enabled to pass all curvatures with the facility of two simple wagons connected in the ordinary manner.

No attempt has yet been made to accommodate the locomotive carriage to the passage of curvatures, by providing the means of changing the parallelism of the axles, and giving them the relative inclination that the radius of curvature requires.

As a result of this communication the company authorized the construction of several "steam carriages" on this plan. Mr. Allen accordingly left Charleston early in the summer of 1831 for the North, and contracted with the West Point Foundry for the construction of the engines. The first one built and put in operation was the *South Carolina*. She was received at Charleston in January, 1832, and was put in operation in February, 1832. Three others were also constructed and put in operation before the end of 1832.

On October 1, 1834, a patent was granted to Ross Winans, of Baltimore, for eight-wheeled cars with two trucks. In his specification he described his invention as follows:

I construct two bearing carriages, each with four wheels, which are to sustain

the body of the passenger or other car, by placing one of them at or near each end of it, in a way to be presently described. The two wheels on either side of these carriages are to be placed very near to each other; the spaces between their flanges need be no greater than is necessary to prevent their contact with each other.

* * * * * * * * *

The body of the passenger or other car I make of double the ordinary length of those which run on four wheels, and capable of carrying double their load.

This body I place so as to rest its whole weight upon two upper bolsters of the two before-mentioned bearing carriages or running gear.

The Newcastle & Frenchtown Turnpike & Railroad Company built or used cars containing Winans's improvements, but denied the validity of his patent. Hence, in 1838, Winans brought his first suit at law against that company. This was the beginning of twenty years of litigation with the railroads of the country, a brief history of which was written by Mr. William Whiting, of Boston, counsel for some of the defendants, and published in 1860 with the title "Twenty Years' War against the Railroads." Of this litigation Mr. Whiting said:

It was at one time a question of millions, to be assured by a verdict of a jury—not indeed in a single suit, but as the result of enforcing the plaintiff's claim wherever railroads were in use and the courts of the United States had jurisdiction. A single verdict, sustained by the court, would enable that result to be easily reached. Stimulated by such hopes and fears, the litigation has been conducted with a corresponding perseverance, labor, and talent. From Maine to Maryland, through a period of eighteen years, in various courts of law and equity, against a great number of railroad companies and against other defendants, before juries of the country and juries of the city, before not less than six different judges of the courts of the United States, and with all the talent and learning that abundant means and a liberal hand could supply, with a pertinacity of purpose rarely equalled, the plaintiff has pressed his claims.

The case was finally carried to the Supreme Court in Washington and was heard in 1858, and a final decision was given in favor of the railroads and against Winans. In closing his account of this remarkable trial Mr. Whiting said:

Thus, after *twenty years* of controversy—the commencement of a large number of actions at law and in equity against the railroads, the actual trial of eight cases—the recovery, by Mr. Winans, of two verdicts sustaining his patent, the disagreement of the jury in three trials, the finding of one verdict at Canandaigua against him, the expenditure, on both sides, of not less than $200,000, and the authoritative settlement of the suit at Washington in favor of the railroads, ends this remarkable chapter in the history of litigation.

In these trials the testimony of Mr. Allen, and the eight-wheeled engines which he built in 1831, became very important evidence, and in an opinion given by Judge Nelson he said:

"The decided preponderance of the evidence is, that this steam carriage embraces all the elements, arrangements, and organization to be found in the cars manufactured by the defendant." *

* * * * * *

The drawings of these engines were obviously imperfect in many respects, but they are all that is left as a record of the construction of these remarkable machines. They show how, at that early date, Mr. Allen anticipated what have since been recognized as essential principles in the construction of locomotives and the operation of railroads. These principles are, first, vertical, and second, lateral flexibility of wheel-base, and third, the distribution of weight of the locomotive on more points and over a greater surface of the road than was possible with the engines in use previous to 1830. These results, it will be observed, are a consequence of the adoption of the swivelling-truck in locomotive construction, which was devised by Mr. Allen in 1830, and which he constructed in 1831 and put in practice in 1832.

Furthermore, as Judge Nelson remarked in giving an opinion in the Winans case, these engines "embraced all the elements, arrangements, and organization to be found in cars with two four-wheel trucks." It was the early adoption of swivelling-trucks for both locomotives and cars in this country which has so materially "differentiated" American railroad practice from that in other countries, and to Mr. Allen belongs the credit of having had the prescience to see, and the courage to put in practice, what are now recognized as essential principles in railroad construction. With the light and experience of sixty years to guide us, it is now easy to see how very close these early engines of Mr. Allen came to being a most brilliant success. If the driving-axles had been attached to frames fastened to the fire-box, and if the two pairs of small wheels alone had been connected to the truck-frames, and the cranks had then been placed at right angles to each other, and the driving-wheels coupled together, the engines would have achieved immortality.

In 1834, after the South Carolina Railroad was finished, Mr. Allen married Miss Mary Moncrief Simons, daughter of the

* Several pages of Mr. Forney's original paper are omitted at this point, bearing on the question of flexible wheel-base.—*Secretary.*

Rev. James Dewar Simons, Rector of St. Philip's Church in Charleston. He remained in Charleston until 1835, and in the spring of that year he went abroad, accompanied by Mrs. Allen and her mother, and devoted nearly three years to foreign travel, returning to America near the close of 1837. After a summer in England and Scotland and a winter in Paris, he visited the principal cities of the continent, and made the entire passage of the Danube to the Black Sea and Constantinople, went thence to Smyrna, the Asiatic coast, to Athens, and across the Levant to Alexandria, and spent the winter on the Nile. In the spring of 1837 he went to Naples and Rome, and returned to Paris, and from there to England, from which he sailed for New York late in 1837.

In 1838 he received the appointment of principal Assistant Engineer of the Croton Aqueduct, John B. Jervis being the Chief Engineer. Before the High Bridge over the Harlem River was built, Mr. Allen recommended that the Croton Aqueduct be carried in a tunnel below the river. Since then this plan has been adopted for the new aqueduct, which now passes under the river. On the completion of the Croton Aqueduct, in 1842, he first turned the water on to supply the city of New York. Afterward he was appointed one of the five commissioners who were intrusted with the supervision of the distribution of the water through the city.

About 1842 Mr. Allen became one of the proprietors of the celebrated Novelty Works in New York. This establishment originated in a somewhat curious way. Previous to the date when this firm was organized, Dr. Nott, then President of Union College, invented a stove and a steam boiler for burning anthracite coal. To show the entire practicability of his invention he had a small steamboat built, called the *Novelty*, which ran from New York to Harlem. At night this boat was laid up at a landing at the foot of Twelfth Street. A small shed was then erected there, with a few tools for doing repairs on the boat. This shop was extended, and it came to be known as "The Novelty's Works," and afterward passed into the hands of Mr. Stillman, who extended it and did various kinds of machine work there. After Mr. Allen entered the firm the business grew very rapidly in various directions, and included mill-work of various kinds, stationary and marine engines, pumps, sugar machinery, steam fire-engines, hydraulic presses, etc.

The firm of Stillman, Allen & Co. was formed in 1847, and consisted of Thomas B. Stillman, Horatio Allen, Robert M. Stratton, George F. Allen, and William B. Brown. The Novelty Works finally became the largest establishment in the country for building marine engines. The machinery for many of the old Collins line of steamers and the Pacific Mail Steamship Company was built there, including the engines of such ships as the *Pacific*, *Atlantic*, *Adriatic*, *Arctic*, and *Baltic*. All these had side-lever engines excepting the *Adriatic*, which had oscillating engines, with two cylinders 9 or 10 feet diameter and 13 feet stroke.

Mr Allen was always a great advocate of oscillating cylinder engines for side-wheel steamships, and in 1867 he wrote what he called "a statement of facts and considerations in reference to beam and oscillating engines for marine side-wheel steamships," which was addressed to Allan McLane, President of the Pacific Mail Steamship Company, and was afterward published in a pamphlet. In this Mr. Allen compared engines with 85-inch cylinders and 8 feet stroke, and claimed that the room occupied by the oscillating engine compared with the beam engine is 8,500 cubic feet for the one and 14,750 for the other; the weight 138 and 152 tons respectively; the number of parts through which the power is transmitted from the piston to the crank is three for the oscillating engine and nine for the beam engine; the number of parts which must be constructed in true line and relation to each other is three for the oscillating and six for the beam engine; the number of bearings and their brasses to be kept in proper adjustment and lubrication is five for the oscillating and thirteen for the beam engine. He also explained that the structure through which the power was transmitted from the cylinder to the crank-pin would be much stronger and more substantial, and the strain on the bottom of the vessel less, with the oscillating than with the beam engine; the weight which comes to a state of rest in passing the centres is $6\frac{1}{2}$ tons in the one engine and 30 in the other; the valve-gear of the oscillating engine, it was admitted, has more parts than that of the beam engine. It was also claimed that all the journals of the oscillator are as accessible as those of the beam engine; that the first could be balanced by cast-iron buckets on the wheel as perfectly as the latter is by the weights at each end of the beam.

The *Adriatic* had oscillating engines, and Mr. Allen applied

his two-motion cone valve-gear to them, which consisted of large conical plug-valves. They were moved by a mechanism which first lifted the valves and then turned them, which it was supposed would prevent them from jamming. There was considerable difficulty in getting them to work satisfactorily, which caused much delay and expense. Some part of the gear broke down on a trial trip, and the valves were finally taken out and others were substituted for them. As there was great rivalry at that time between the Cunard line which was owned by Englishmen, and the Collins steamers which were owned by Americans, this experimental valve-gear attracted a great deal of attention, and was the subject of much criticism.

The engines for the *Constitution*, *Moses Taylor*, *Ancon*, *Mariposa*, *Great Republic*, *Idaho*, *Montana*, *Arizona*, *Golden Age*, and *Golden Gate* for the Pacific Mail Steamship Company, and for the *Re d'Italia* and *Re Galantuomo*, two war-ships for the Italian government, were all built at the Novelty Works. The *Golden Gate* also had oscillating engines.

During the war, engines were built for three gun-boats and also for the sloops *Adirondack* and *Wampanoag*, and the double-turreted monitor *Miantonomah*, and the frigate *Roanoke* was converted into a monitor with two turrets. The old engines were used in the vessel when it was altered.

At one time there were over 1,500 men employed in the Novelty Works, but so great was the difficulty of getting men at that time, owing to the demands of the army and navy, that Mr. Allen went to Europe and employed a large number there, who were brought over.

It was during the war that the somewhat acrimonious dispute with reference to the economy of using steam expansively arose. On the one side were those, including Mr. Isherwood, the Chief of the Bureau of Steam Engineering, who advocated the use of moderate degrees of expansion and comparatively low pressure with the engines then in use; and on the other side was the late Mr. E. N. Dickerson and others, who claimed that greater pressures and excessively high degrees of expansion were the most economical. The subject was discussed on both sides with great fierceness and attracted the attention of Congress, and finally the Naval Committee requested the Naval Department to have a series of experiments made "to assist in determining the limitation of the economical expansion of steam under practical con-

ditions and other collateral questions relating to the general subject." *

In 1863 Mr. Welles, then the Secretary of the Navy, appointed Mr. Allen and Mr. Isherwood, who at that time was the Chief of the Bureau of Steam Engineering, "a commission to devise and conduct a set of experiments to ascertain, by means of practical results, the relative economy of using steam with different measures of expansion." † The commission was also authorized "to associate such other persons as it may deem advisable for the object in view."

The experiments were to be made under the supervision of a committee of the Franklin Institute of Philadelphia and the Smithsonian Institute of Washington, and three civilian engineers, of whom Mr. Allen, who believed in high rate of expansion, was one. If committees were appointed by the institutions named, they did not take an active part in conducting the experiments. These were made during the years 1864 to 1868 at the Novelty Works, under the general direction of Mr. Allen, who was then the president of the company, and Chief Engineer Isherwood, at that time the Chief of the Bureau of Steam Engineering, U. S. N., who detailed a corps of assistants to do the work.

The experiments were commenced at the Novelty Works on a large scale. Engines with various-sized cylinders were constructed. These were connected with a large air-fan, the revolutions of which represented the work done. Much time and money was consumed in getting this machinery to work satisfactorily, and in making the experiments, and apparently they did not prove exactly what either side anticipated. While they were insufficient to settle all the points at issue, they showed what is well known now, that the point of cut-off which is most economical becomes shorter as the pressure is increased; but that with any pressure, the most economical degree of expansion is soon reached, and the cost rises rapidly after this point is passed.

So much time was consumed in making experiments that some of those who formed the commission lost interest and practically

* Paper on the Cost of Power in Non-Condensing Steam-Engines. Read by Charles E. Emery before the American Society of Mechanical Engineers, 1888.

† The above language is quoted from the original appointment of the commission.

abandoned them, possibly because the results did not prove what it was expected they would. The work was then carried on by Chief Engineer Isherwood and his assistants in consultation with Mr. Allen. The cost of the experiments went up to over $100,000, and as the time consumed was so great and the results were apparently inconclusive, the Navy Department finally ordered them discontinued. The commission in charge of them never made a report nor were the results published under government authority, although a general table was furnished by Mr. Isherwood to Mr. R. H. Buel, who had it published in the articles on steam engineering which he prepared for Appletons' "Cyclopædia of Mechanics" and in the American edition of "Wiesbach's Mechanics."

It is not easy now to learn what was the precise significance, or rather what was proved by these experiments. Apparently they did not show as great an economy from the use of high-pressure steam and high rates of expansion as the advocates of that side of the question expected, though the results with high-pressure steam showed greater economy than those with low-pressure, and it was also found that it was economical to cut off shorter with high than with low pressure steam. The experiments were, of course, made with engines of the kind then in most general use, and did not include compound or triple-expansion engines, with the very high pressures which have been made practicable by their use. It is evident now that those who then advocated the use of high-pressure steam and excessively high degrees of expansion did not understand fully how steam used under these conditions is affected by various circumstances, especially those existing when steam is expanded in a single-cylinder engine.

Afterward, a competitive trial was made by Commodore Isherwood and Mr. Dickerson with two United States vessels, the *Winooski* and the *Algonquin*. They were first tied up in a dock, and their wheels were turned at a regular rate and a careful record was kept of the fuel consumed. In these trials the *Winooski*, Commodore Isherwood's vessel, had an engine with double poppet-valves and Stevens cut-off. The *Algonquin* had a Sickles cut-off with single poppet-valves.

Trials of speed were afterward made at sea. The *Winooski* then used steam of 25 lbs. pressure cut off at $\frac{6}{10}$ of the stroke, and the *Algonquin* carried 90 lbs. of steam and cut off at $\frac{1}{10}$.

The failure of the *Algonquin* in these trials is now a matter of history.

After the experiments at the Novelty Works were ended, Mr. Charles E. Emery, who was an assistant engineer in making them, suggested a supplementary series with a small engine having 8 x 8 inch cylinders. The officers of the Novelty Works agreed to bear the expense of these, which amounted to about $5,000. The results of these experiments with non-condensing engines were afterward published by Mr. Emery in the proceedings of the American Society of Mechanical Engineers in 1886 and 1888.

While the investigations were being made it is said that Mr. Allen suspended judgment thereon, as was proper he should. Those who were intimately associated with him at the time never heard him express an opinion with reference to the subject after the experiments were ended.

In the light of our present knowledge it seems singular that experiments on such a scale were needed to show what now seems so easily proved. Doubtless some of the experiments of the present day will appear equally needless twenty-five or thirty years hence.

During all of Mr. Allen's career he was a prolific inventor, as will appear from the following list of some of the patents which he took out:

Steam Cut-Off. H. Allen	1841
Stop-Cock. "	1841
Steam Cut-off. "	1842
Determining Thickness of Metal Pipes. H. Allen	1843
Tapping Mains. H. Allen	1843
Steam Cut-Off. "	1847
" " "	1849
" " "	1849
Steam-Engine Valve-Gear. Allen & Wells	1853
" " " " " "	1853
Two Motion Cone-Valve	1855
Steam Engine Valve-gear. H. Allen	1857
Steam Boiler Tube Joint. "	1858
Car Seats and Couches. "	1866
Connecting the Tubes with the Heads of Surface Condensers. H. Allen	1868
Sleeping Cars. H. Allen	1876
Terrestrial Globes "	1879

The Allen & Wells cut-off, in its several different forms, was introduced to some extent, and is still in use on different steam-

boats. The method of connecting condenser tubes to their heads with compressed wooden ferrules has also been extensively adopted.

During the war, although a great deal of work had been done at the Novelty Works, the success of the company was not proportionate to the amount of the business. They were operated during part of that period under the disadvantage of a market in which the prices of labor and materials were constantly rising. Contracts were taken at fixed sums, and it was then not easy to anticipate what the increase would be in the cost of doing work before it was finished.

When the war was ended there was, of course, a cessation of government work. Business at the Novelty Works had been conducted on a large scale, with fixed expenses in the same proportion. The tools and machinery were old and out of date, and it was soon found that the works were being conducted at a loss. To remodel and reëquip them with new tools and machinery to meet the changed condition of business would involve a large outlay of capital. The real estate was very valuable, and it was finally determined to close the works and wind up the business. This was done in 1870, and the Novelty Works soon ceased to exist. The business which was conducted there, like most great enterprises, was attended with varying success. Under the firm of Stillman, Allen & Co. it was at first very profitable, but some heavy losses embarrassed the firm and they had to seek outside aid. Mr. James Brown furnished the firm with more capital, and when a stock company was organized he became a stockholder, and Mr. Allen was president. During the war the business was very active and during part of the time profitable, but Mr. Allen was then not a large holder of the stock.

During his connection with the Novelty Works he also acted in the capacity of Consulting Engineer for the Erie Railroad, and he was President and Chief Engineer of that company for a year. He was also Consulting Engineer to the Panama Railroad Company for a short time, and during that period also held incidentally other important engineering trusts. His professional career may be said to have ended as Consulting Engineer of the Brooklyn Bridge.

In 1870 Mr. Allen ıetired from active life and built himself a house at Montrose, near Mountain Station on the Morris &

Essex Railroad, in New Jersey, where here sided up to the end of his life. He left a widow, three daughters, and a son. He always seemed to derive his chief enjoyment in life from his delightful home, but this was especially the case during the latter years of his life. He was a man of very quiet domestic tastes, but took a lively interest in engineering, scientific, and especially educational matters, up to the last.

He always took an active interest in philanthropic and charitable matters, and was one of the founders of the Union League Club, and an active member of it in the days when its influence was exerted in behalf of great national questions, and before narrow partisanship had contracted its sphere of usefulness. He was also one of the organizers, and for a long time an active member, of the Association for the Improvement of the Condition of the Poor, the Children's Aid Society, and the New York Gallery of Art, and was associated with a number of gentlemen who were instrumental in preserving what was known as the Abbott Collection of Egyptian Antiquities, which now forms a part of the New York Historical Society's collection. He was a member of the American Society of Civil Engineers, and was its president for one term. He was also a member of the American Society of Mechanical Engineers, and was elected an honorary member of both societies.*

Mr. Allen took an interest in a very wide range of subjects. During his later years he devoted much time to the subject of education. He earnestly desired to be of use to the rising generation, and he sympathized very strongly with the difficulties of children and young people in acquiring knowledge. He wrote a little book on arithmetic and commenced one on algebra, and was especially interested in the methods of teaching astronomy. He also wrote a book on that subject, and invented and constructed a number of instruments to facilitate the study of astronomy in schools.

His life and experience, if it could be fully written, would be of exceeding interest.

In his later years he often expressed regret that he did not keep a record of the events of his early life, and especially his observations during the period that he first visited Europe. He was then on intimate terms with George Stephenson and

* Mr. Allen was the first honorary member elected to the latter society after its formation.—*Secretary.*

the early fathers of the railroad system. He was in England to study that system, which was then, if not in its infancy, at any rate in its early youth. If his observations had been fully recorded they would now be of intense interest. Beginning his study of engineering in early manhood, when railroads were an experiment, it extended over the period, so recently ended, which covered completely that wonderful era of modern development which has been due to the introduction, application, and diffusion of steam-power over the whole civilized world.

Mr. Allen was an ardent lover of nature, and nearly always devoted the early morning hours to the enjoyment of its beauty and took the keenest delight in its contemplation. Among the marked traits of his character were his gentleness and generosity, which, it is said, "is in nothing more seen than in a candid estimation of other men's virtues and good qualities." He was always ready to give a helping hand to those who were down and trying to get up. His words and acts of encouragement to many young men beginning the hazardous voyage of life were like propitious breezes and inspired them with hope, which sustained them until they reached port. A paper published near his home said of him: "His integrity was of the most unswerving, unflinching kind, and he was scrupulous almost to a fault over matters that ordinarily pass current in the mercantile world." The modern forms which business bribery has assumed excited in him unbounded indignation. A gentleman occupying a prominent position in public life, and who was associated with Mr. Allen during the trying period of the war, said of him he possessed all the best qualities of a true gentleman.

His last years were spent quietly with his family in his home at Montrose, in New Jersey. It may be said of him, that his integrity commanded the respect of all honest men who knew him, his generosity made many persons his debtors, and the delight which he took in contributing to the happiness of others led all to be "kindly affectioned" to him.

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